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Boiler Stack Gas Heat Recovery

ABSTRACT This report is a detailed study of various heat recovery schemes for Navy shore facilities to utilize otherwise lost stack heat. The waste heat can be used alternatively to improve the boiler efficiency through feedwater and/or combustion-air preheating, or to lighten the boiler load through process-steam/water heating. The procedure for estimating the energy potential of a given flue-gas stream is explained in detail with sample plots. Based upon economic analysis of available options, three cost-effective methods of recovering waste heat were identified: the conventional economizer, the direct-contact heat exchanger, and the indirect-contact condensing heat exchanger with temperature-raising by heat pumps, if appropriate. Important items such as materials, corrosion, maintenance, control, and retrofit are discussed. It is recommended that the entire process of selection and planning be incorporated in Computer Aided Engineering (CAE) software, and the conclusions be verified by testing in an actual retrofit unit.
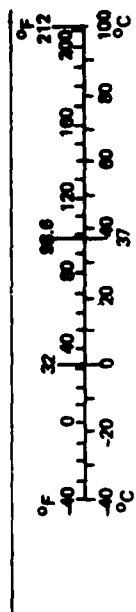
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METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2,000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in = 2.54 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286. Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1,000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

°F -40   0   32 40   80   98.6 120   160   200 212
°C -40   -20   0   20   37   40   60   80   100

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CONTENTS

A-1

INTRODUCTION

Flue gas from a boiler represents a sizable source of recoverable energy. Success in recovering some of this energy would bring about direct and continual savings. Table 1 shows the size distribution, extent of operation, and annual fuel usage of Navy stationary boilers. It shows that large boilers represent only 12.4 percent of the total boiler population, but they are responsible for about 80 percent of the total fuels consumed by all the boilers. It is, therefore, most beneficial for the Navy to consider energy recovery projects for such large boilers. With boiler stack loss conservatively estimated at 17 percent of the fuel usage, these boilers discharged approximately 8.4 trillion Btu per year as enthalpy of flue gas during FY85. It is reasonable to expect that half of the flue gas energy can be recovered, which is equivalent to a saving of at least 30 million gallons of oil used annually by the Navy and Marine Corps. This saving is significant especially in view of reduced fuel oil dependence of Navy shore activities.

This report is a summary of the current technology on waste heat recovery, practical approaches to boiler efficiency improvement, and other concepts for utilizing the recovered energy from the flue gas. Practical suggestions for estimating the energy potentials of flue gases, mode of energy recovery, over-all equipment size and payback period, and selecting among alternative designs are included.

On an average, the flue gas temperature leaving Navy shore facility boilers is about 400 °F, ranging between 350 °F and 650 °F. Flue gas leaves the boiler at a temperature higher than the steam temperature for heat transfer to take place, and to preclude condensation in the stack. Figure 1 illustrates typical flue gas temperatures above steam temperatures at various boiler firing rates with no more than 20 percent excess air.

1

Boiler efficiency is defined as:

$$\eta_{boiler} = 1 - \frac{\Sigma \text{ (Energy Losses)}}{\text{(Energy Released by Fuel Combustion)}}$$

It is seen that any decrease in energy losses is tantamount to a higher efficiency and reduced fuel consumption.

Typical boiler efficiency ranges from 75 to 85 percent. It can be improved by first identifying the nature of various energy losses and their relative magnitudes, and then reducing the dominant losses responsible for the low efficiency. A representative breakdown of various boiler losses is shown in Figure 2.

The loss due to radiation and convection across the insulation of the boiler is essentially constant, largely independent of the boiler firing rate. It thus becomes a larger percentage of the total energy loss at lower boiler loads (Ref 1). At higher loads, this loss is usually 1 to 2 percent of the energy input to the boiler.

Another energy loss associated with a boiler is caused by incomplete combustion of the fuel. It is normally caused by poor mixing or inadequate air supply. This loss can be minimized through burner design or adjustment, and proper maintenance or control. For a well tuned boiler, this loss is limited to about 0.5 percent of the energy input.

Blowdown is necessary in order to control dissolved solids in boiler water. Typically, blowdown accounts for about 1 to 3 percent of the boiler energy input. Commercial equipment is available to recover some of this lost energy.

By far the largest energy loss in a boiler is attributable to the hot flue gas leaving the stack. Thus, the most effective method to save fuel is to recover as much sensible and latent heat of water vapor condensation in the flue gas as practicable. Additional benefits of this energy recovery are:

1. An increase in stack gas density, which will permit a smaller stack diameter.

2. Some enhancement in the combustion efficiency, which may reduce the amount of excess air needed at the burner which, in turn, will reduce the amount of stack gas and pollutant emissions.

3. The higher density of the cooled stack gas increases the concentration of pollutants and makes their removal easier.

4. The condensation of H_2SO_x promoted by a colder stack gas reduces SO_x emission into the atmosphere.

An important factor influencing stack gas heat recovery is the corrosion problem accompanying the cooling of the gas. Because the sulfuric acid dewpoint is higher than the water vapor dewpoint, latent heat recovery means condensate will contain sulfuric acid. To avoid the consequential corrosive effect, traditional practice has been to limit heat recovery to a minimum stack gas exit temperature of around 350 °F. Within the last decade, however, corrosion-resistant materials have been developed and applied to commercial heat recovery systems, many of which successfully operate below the acid dewpoint. One of the objectives of this study is to demonstrate that the technology base now exists which allows maximum recovery of the thermal energy in stack gas while keeping the corrosion problem under control.

Evaluation of the energy recovery benefits of this approach needs to take into consideration the following trade-off items:

1. Equipment and installation costs

2. Added maintenance

3. Additional power requirements for fans, pumps, etc.

4. Corrosion, erosion, and fouling of the heat exchanger

5. New assessment of total environmental impact

6. Ease of retrofitting

The main text of this report is organized in the following manner. To begin with, the procedure for estimating the energy potential in a flue gas is explained, followed by discussion of such essential parameters as hours of operation, boiler size and load, operating load schedule, stack gas temperature, excess air, makeup feedwater, and fuel type. Economic considerations are then focused on the life cycle savings. A general survey of heat recovery devices then covers heat pipe, plate type heat exchanger, heat wheel, run-around coil, direct contact heat

3

exchanger, economizer, indirect-contact condensing heat exchanger, organic Rankine cycle, and heat pump, including their basic characteristics, merits and demerits. Next, using a parametric evaluation technique, economizers and the indirect-contact condensing heat exchangers have been identified as the two most suitable candidates for waste-heat recovery via boiler-efficiency improvement. For each of these two approaches, a general discussion is presented covering such items as system operation, corrosion, maintenance, materials, controls, retrofit, load to be used in sizing, and others. A third scheme using a combustion-air preheater either by itself, or in combination with either of the previous two schemes is also discussed. Another parametric evaluation identifies the indirect- and direct-contact heat exchangers as favorable processes for heating applications. To provide some experience-based guidelines for the planner, eight actual cases gathered from various sources are described. The report ends with conclusions, and recommendations for future work.

ENERGY IN STACK GAS

When considering heat recovery methods, it is essential that the quantity and quality of the heat source be identified. Once the amount and temperature level of the heat source are established, the feasibility of applying various heat recovery schemes can be studied. Depending on the mode of heat recovery, the amount of recoverable energy, hardware requirements, system efficiency improvements, etc. can be quantitatively determined.

Since the boiler stack gas is the heat source here, a simple combustion analysis is necessary as the first step toward an estimation of the energy potential of the gas.

Combustion Analysis

Among the elements in various fuels, hydrogen and carbon are the major combustibles. Another common (although undesirable) combustible element is sulfur. A fuel frequently also contains some oxygen which, when released, serves as a part of the oxygen required for combustion.

Based on simple chemical reactions, and denoting by C, H, S, and O the respective mass of each species in lbm/lbm fuel, the theoretical oxygen required for combustion (lbm/lbm of fuel) is given by:

$$\text{Theoretical Oxygen} = 2.67 \cdot C + 7.94 \cdot H + S - O$$

Since air contains 23.2 percent of oxygen by weight, 4.31 lbm of air is needed to supply 1 lbm oxygen, and we can write

$$\text{Theoretical Air} = 11.51 \cdot C + 34.2 \cdot H + 4.31 \cdot S - 4.31 \cdot O$$

All boilers operate at certain levels of excess air to assure complete combustion. This additional air must be supplied, and accounted for in the products of combustion. For a given percentage of excess air, ε, we have

$$\text{Total Air} = (1 + \varepsilon)(11.51 \cdot C + 34.2 \cdot H + 4.31 \cdot S - 4.31 \cdot O)$$

The products of complete combustion thus contain excess oxygen given by:

$$\text{Excess } O_2 = \varepsilon \text{ (Theoretical Oxygen)} = \varepsilon (2.67 \cdot C + 7.94 \cdot H + S - O)$$

Other products of complete combustion are:

CO_2 --- $(C + 2.67 \cdot C) = 3.67 \cdot C$

SO_2 --- $2 \cdot S$

H_2O formed by combustion of H_2 --- $8.94 \cdot H$

H_2O in air supply --- $M \cdot \text{(Total Air)}$

H_2O in fuel

5

N_2 contained in air --- $0.7685 \cdot (\text{Total Air})$

N_2 in fuel

Ash

where M is the moisture in the combustion air (lbm/lbm dry air), and all other items are in lbm/lbm fuel. The total mass of flue gas produced per lbm of fuel is obtained by summing up the items for excess O_2, CO_2, SO_2, H_2O, and N_2 as listed. Figure 3 shows components of air supply, fuel and combustion products.

Calculation of Dew Point of Flue Gas

The dew point of the flue gas is the saturation temperature of the water vapor at its partial pressure. It indicates the temperature at which water vapor within the flue gas begins to condense.

Denoting by CO_2, N_2, etc. the species of the flue gas in lbm/lbm of fuel, each species may be converted to a molar basis by simply dividing by its molecular weight. The mole fraction of water vapor in the flue gas can then be calculated as

$$n_{H_2O} = \frac{(\text{Total } H_2O)/18.02}{\dfrac{CO_2}{44.01} + \dfrac{SO_2}{64.07} + \dfrac{O_2}{32} + \dfrac{\text{Total } N_2}{28.02} + \dfrac{\text{Total } H_2O}{18.02}}$$

The partial pressure of water vapor, P_{H_2O}, in the flue gas is then given by

$$P_{H_2O} = n_{H_2O} \cdot P_{\text{flue gas}}$$

where

$$P_{\text{flue gas}} \simeq P_{\text{atm}}$$

The dew point, or the saturation temperature at P_{H_2O} may then be found from the saturated steam table. Figure 4 was prepared in this way for the reader's convenience. The reader may wish to take advantage of

6

a steam-table software, called WTSTM(87), developed at NCEL (see NCEL
TM: 73-86-24 WTSTM87 - A Water/Steam Table Software Based on IAPS-1984
Formulation, Including the Two-Phase Region, by P. C. Lu and T. T. Fu.).

The dew point discussed here is often referred to as the "initial
dew point" to signify the threshold of initial condensation of water.
It should not be confused with the "acid dew point" of the flue gas,
which indicates the initial formation of liquid H_2SO_x. The acid dew
point is much higher than the initial dew point.

Calculation of the Flue-Gas Energy

The total energy in a flue gas stream is the sum of the enthalpies
of its components. The reference condition used for the calculation is
fixed conventionally at 77 °F and 1 Atm.

Water vapor in the flue gas is in a superheated state above the
initial dew point. The cooling process and enthalpy change of water
vapor may be illustrated on the Mollier diagram as shown in Figure 5.
The cooling first reaches the initial dew point at which condensation
begins. Further cooling is effected by following the saturation line
with more and more water vapor condensing. The partial pressure of
water vapor decreases continually, accompanied by a reduction in dew
point. Therefore, the stack gas temperature must continuously be low-
ered to condense more water vapor. As water vapor is condensed, heat of
condensation is released.

The mole fractions of species in the flue gas are ascertained first
for a given fuel composition and a prescribed excess air. The initial
dew point is also determined before starting the procedure. The enthalpy
difference of the flue gas between any temperature T (larger than the
initial dew point, and smaller or equal to the boiler exit temperature)
and the initial dew point, T_d, is calculated by integrating from T_d to T
the expression $C_p(T)dT$, with the specific heat, C_p, weighted by the mole
fraction of the species, and summing over all species. Starting with T_d
and going downward, one assumes that a certain small percentage (say
10%) of the original water vapor content is instantaneously condensed
into liquid. This assumed mass of condensation triggers a drop in the

7

mole fraction of water vapor, and hence in the partial pressure of water vapor. The saturated steam table then yields a lower dew point $T_{d'}$. The enthalpy of the flue gas at $T_{d'}$ is now lower than that at T_d for two reasons: the mass of condensation times the heat of condensation, and the summation over all species of integrations, from $T_{d'}$ to T_d, of $C_p(T)dT$ weighted by the new mole fractions. This procedure is repeated from $T_{d'}$ downward until 77 °F is reached. (Care should be exercised in the datum value of enthalpy according to the saturated steam table.) The final result is then the total energy in the flue gas, in Btu/lbm fuel, at all temperatures, with a zero-value assigned to T = 77 °F.

The inefficiency of a boiler due to stack losses may be represented by the ratio of the flue gas enthalpy to the high heating value (HHV) of the fuel. Figure 6 illustrates results of such calculations as a function of flue gas temperature at several levels of excess air when burning natural gas. It is clear that the energy contained in the flue gas is too significant to be discarded. The enthalpy of the flue gas (in Btu/lbm fuel) at a specific temperature can be determined by multiplying the corresponding ratio by the high heating value of the fuel. The sudden breaks in slope in the figure indicate initial dew points. Roughly half of the energy within the flue gas is in the form of latent heat, the exact fraction being dependent on the fuel type. Sensible heat can be recovered at higher (and, therefore, more useful) temperature levels than latent heat. However, the amount of energy available per degree drop in flue gas temperature is much greater when recovering latent heat below the initial dew point. The effect of increased excess combustion air is increased loss of sensible heat via the stack gas.

Increased excess air tends to decrease the dew point since it decreases the mole fraction of water vapor in the stack gas. This can be seen more clearly in Figure 7 which shows only the condensing portion of Figure 6. The effect of excess air on flue gas enthalpy below the initial dew point is small because of the domination of the latent-energy loss.

Figure 8 is a composite plot for various fuels. It shows that natural gas boilers have the highest stack losses. This is due to the higher amount of hydrogen in natural gas (see Table 2). For every lbm of hydrogen in the fuel, about 9 lbm of superheated water vapor is formed in the combustion products. Presence of hydrogen in the fuel has a much greater

8

effect on the amount of latent heat when compared to moisture present in the air or fuel. As an example, coal-water slurry contains about 30 percent water and only about 3.5 percent hydrogen. As illustrated in Figure 8, flue gas from firing coal-water slurry does not carry away as much heat as that from firing natural gas. Similarity, a coal-fired boiler has the lowest stack loss since coal contains only about 4.8 percent hydrogen.

Summary

1. The amount of energy in the stack gas is too significant to be ignored.

2. Boilers burning natural gas or coal-water slurries are the most likely candidates for stack gas heat recovery; those using coal are the least likely candidates.

3. Latent heat can account for about one-half of the total energy in the stack gas. Its recovery, however, can be accomplished only at low temperatures.

4. While the amount of latent heat in the stack gas may look attractive, its recovery is hindered by the low temperature level at which the recovered latent heat is available. (There is inherent inefficiency associated with recovering low temperature energy.)

5. As excess air is increased, the water vapor dew point decreases and sensible stack gas heat losses increase.

6. Both the stack loss and its latent energy component are increased very considerably by the hydrogen content of the fuel.

7. Sensible energy can be recovered at relatively high temperatures as compared to latent energy. However, for the same change in stack gas temperature, much greater quantities of energy are available below the initial water vapor dew point rather than above it.

BASIC CONSIDERATIONS

Before heat recovery alternatives can be considered, overall boiler information is needed to determine the amount of energy currently being lost in the stack gas, fraction of this energy that is potentially recoverable, promising heat recovery schemes, and the economic feasibility of heat recovery. The various factors affecting heat recovery from boiler stack gases are discussed below.

Hours of Operation

The number of boiler operating hours is, by far, the most important factor needed to justify the installation of a heat recovery system. For a given boiler operating at a set load, the energy savings increase and the payback period decreases directly with an increase in the number of operating hours, regardless of the type of fuel and heat recovery scheme being adopted.

Boiler Size and Load

The installation of a heat recovery system generally tends to be economically more attractive for boilers of larger sizes and/or those operating at higher loads. Increased boiler loading and larger boiler size means greater mass flow rates, and therefore greater quantities of stack gas from which energy could be recovered. At the same time, the installed cost of a heat recovery system increases more slowly with its size because of the basic costs associated with the installation regardless of size (structural support framework, piping, ductwork, etc.). Thus, for larger boilers or boilers operating at higher loads, a higher energy-savings-to-capital-investment ratio can be realized.

Operating Load Schedule

Throughout a year, a boiler will operate at different loads during different periods. During the winter months the boiler may operate at 75 percent rated load 100 percent of the time, but during the summer the

10

boiler may operate at only 25 percent rated capacity 50 percent of the time. The total energy recovered on a yearly basis can be calculated by summing up the energy saved during different periods at specific load conditions.

Stack Gas Temperature

Boilers are designed to operate at a particular flue gas temperature. Various factors may affect the flue gas temperature leaving the boiler. For example, deposits on heat transfer surfaces within the boiler will increase the flue gas temperature. Flue gas temperature is an indicator of boiler efficiency. Higher flue gas temperatures signify less efficient boiler operation and good potential for waste heat recovery. A rule of thumb states that boiler efficiency can be improved by 1 percent for every 40 °F drop in flue gas temperature. A higher flue gas temperature corresponds to a greater potential for heat recovery, more efficient heat recovery due to greater temperature differences between the sink and source, and higher temperature level of the recovered energy open to more uses.

Excess Air

All boilers operate with excess air to assure complete combustion and controllability. This excess air contributes to stack loss of a boiler, since a part of the energy released during combustion goes to heat the excess oxygen and nitrogen unnecessarily. Boilers operating at high excess air are good candidates for waste heat recovery.

Makeup Feedwater

Navy shore facilities boilers are commonly used to generate steam or hot water to heat buildings, provide hot water, etc. In steam generating boilers, a portion of the steam returns to the boiler plant as condensate. The temperature of the condensate ranges between 140 and 160 °F. The condensate is supplemented by makeup feedwater. The makeup

11

water enters the deaerator usually at room temperature of between 60 to
80 °F. Boiler efficiency can be improved by preheating the makeup water
using energy recovered from stack gas. Due to the relatively low tempera-
tures involved, boiler makeup water has a much greater potential to absorb
energy than the condensate returning to the boiler. Thus, a large per-
centage of make-up water required by a boiler would make it an ideal
candidate for waste heat recovery.

However, the conventional economizer is not affected in this manner
by the percentage of makeup water, since feedwater entering the economizer
must enter at elevated temperatures to prevent corrosion.

Fuel Type

Combustion products from burning fuels with higher hydrogen content
contain more water vapor and, hence, larger amounts of latent heat. Thus,
gas-fired boilers are inherently less efficient than coal-fired boilers
and represent better candidates for heat recovery. Type of fuel will also
affect the maintainability and service life of a heat recovery system.
Natural gas is a clean-burning fuel and causes minimal corrosion problems
for heat recovery hardware. Fuel oil, however, contains varying amount
of sulfur, which leads to acid corrosion problems. Yet, the relatively
high cost of fuel oil may provide economic justification for a heat re-
covery installation. There are virtually no successful applications of
condensing flue gas recuperators on coal-fired boilers. Ash, sulfur,
etc., in the stack gas from these boilers tend to cause severe corrosion
and fouling. The relatively low cost of coal, and the low latent heat
in the combustion products make heat recovery less attractive.

Economic Considerations

The economics of a heat recovery system is based on its installed
cost, the operation and maintenance costs, price of fuel, service life,
hours of annual operation, and boiler operating parameters. The service
life of most heat recovery systems is estimated to be about 15 years.

Project decisions are often determined in terms of the life cycle
savings to investment ratio. This is defined as follows:

$$\text{Savings to Investment Ratio (SIR)} = \frac{\text{Present Value of Savings Over the Equipment Life}}{\text{Installed Cost of the Equipment}}$$

Present value of savings will depend upon the discount factor and life of the equipment, per NAVFAC P-442. For the purpose of this analysis, a discount rate of 7 percent and a 15-year equipment life was assumed. The Uniform Present Worth discount factor adjusted for average fuel price escalation for gas (Ref 2) with a discount rate of 7 percent is 11.98.

With most boiler heat recovery systems, a smaller and less expensive heat recovery unit could minimize the initial investment. However, the fuel savings over the service life of the unit may also be reduced. On the other hand, for a given boiler operating at a set load, heat exchanger efficiency will level off as the size is increased. So, increasing the size of a heat recovery unit can only command a diminishing return. This additional energy is recovered at the expense of a greater initial investment. Yet, in general, boiler efficiency improvement and SIR will be greater.

The greatest inaccuracy in estimating the economic benefits of a heat recovery system is in predicting its installed cost. In general, the installed heat recovery system cost is between two and five times the hardware cost, depending on the type of heat recovery unit and the difficulty of retrofit. For example, in cold climates, the heat recovery unit will have to be installed indoors to guard against freezing. If no space exists within the boiler room, an added shelter may have to be constructed. Also, if the heat recovery unit is large, installation may require the partial removal of a wall to gain access into a building. Other uncertainties involved with a retrofit installation include costs associated with duct and pipe layout, additional controls, and the updating of the present system.

HEAT RECOVERY DEVICES - A GENERAL SURVEY

The best choice of a heat recovery device is based upon the quantity and quality of the waste heat, intended applications of the recovered energy, and the installed cost of the heat recovery device. Available heat recovery devices are discussed in this section.

Heat Pipe

A heat pipe is composed of a sealed pipe or container partially charged with a liquid (water, freon, ammonia, etc.). It can be divided into two sections: an evaporator and a condenser (see Figure 9). In the evaporator section, heat is absorbed by the internal working fluid by evaporation. The increased pressure at the evaporator end forces the vapor toward the cooler condenser section where heat is removed and the vapor condenses. The internal circumference of the heat pipe is lined with a thin layer of wicking or mesh-type material (felt, screen, foam, or grooves). The condensed fluid migrates by capillary action through the mesh back to the evaporator section and the cycle is repeated. A thermosyphon heat pipe does not utilize a wicking material. Instead, the pipe is placed in a vertical position with the evaporator located below the condenser section. As the vapor condenses, it returns to the evaporator section by gravity. Even with wicking material, it is always advantageous to locate the evaporator section below the condenser so that the condensed fluid is aided by gravity, and does not have to depend solely on capillary action to reach the evaporator section of the heat pipe. Wicks also distribute the liquid uniformly in the evaporator section, separate the vapor from the liquid, and inhibit condensed fluid from becoming entrained in the moving vapor. In cases where the liquid does become entrained in the vapor, dryout is likely to occur at the evaporator section because not enough liquid returns to the evaporator section.

For applications in a stack gas heat recovery system, the design temperature of the working fluid of a heat pipe should be chosen between the flue gas and the cooling air or water temperature so that sufficient radial heat transfer exists at both condenser and evaporator ends. Heat

pipes with liquids at different saturation temperatures can be utilized to obtain optimum system performance for different temperature ranges of the heat source and sink. This is illustrated by a counterflow heat pipe system in Figure 10.

Although heat pipes have the potential to transmit a large amount of energy axially, this energy must be added to the heat pipe at the evaporator end, and removed at the condenser end. Fins are required to increase the heat transfer surface area and radial heat transfer at both ends. Without fins, the heat transfer capability of a heat pipe can be limited. For a condensing stack gas heat recovery system, the evaporator section of a heat pipe will be subjected to a corrosive flue gas environment. Conventional finned metal (aluminum, copper, stainless steel) tubes should not be used unless controls maintain the outer metal temperatures above the sulfuric-acid dewpoint or unless they are coated with a corrosion-resistant material such as an epoxy. Such a coating, uniform and thin, is currently impossible to obtain. Thus, although heat pipes offer a unique and advantageous way to transfer heat, corrosion at the evaporator end can limit its application, reliability, and performance.

Heat pipes are mostly applied to transfer heat from one gas stream to another. The two gas streams must be adjacent to each other because it is often uneconomical to reroute large stream volumes of gas just for the purpose of heat recovery using a heat pipe. Should corrosive conditions occur, causing a tube failure, the individual tube can be removed from operation without shutting down the entire system such as with a conventional tubular heat exchanger. Heat pipes can be compact, operate with low temperature differentials, and have no moving parts to malfunction.

Plate Type Heat Exchanger

Plate-type gas-to-gas heat exchangers can be used in counterflow or crossflow through adjacent passages separated by heat conducting walls (see Figure 11). It is possible to preheat combustion air, process fluids, building makeup air, etc, by this type of heat exchangers. It is a reasonably simple system, but the gas streams must pass in close proximity. Also, for oil or coal-fired boilers, heat recovery in this manner

15

is limited since acid condensation can be a problem. At least one vendor is manufacturing teflon-coated plate heat exchangers to avoid this corrosion problem, and to recover a greater amount of sensible heat as well as a portion of the latent heat, by lowering the stack gas temperature below the sulfuricacid dew point.

Heat Wheel

A heat wheel is a regenerative type heat exchanger which extracts heat from a source, briefly stores it, and then releases it to a cooler sink. It consists of a large rotating wheel-frame packed with a heat absorbing matrix material, a motor drive system, air seals, and a purge section. As the heat wheel rotates, a section of the porous matrix is exposed to the waste-heat gas-stream thereby extracting and storing thermal energy. As this heated section rotates past the cold process air, heat is transferred from the matrix to the air stream. In this manner, the matrix is alternatively heated and cooled, and heat is indirectly transferred from the hot gas stream to the cold air stream (see Figure 12). Cross-contamination between the exhaust gas and supply air streams is minimized by adding a section that continuously purges exhaust gas from the corrugated matrix before being exposed to the supply air. Wheels can exceed 50 feet in diameter. Matrix materials include aluminum, stainless steel, and ceramics for higher temperatures. Disadvantages include cross-contamination of the two streams due to sealing and purging problems, and clogging of the passages which decreases heat transfer, increased pressure drop and motor horsepower. Another disadvantage of this system is that the two streams have to pass in close proximity. Ducting and retrofit costs can make the scheme uneconomical. This unit could be used to preheat combustion air, but not to preheat building air because of contamination. Stack gas from oil and coal-fired boilers will add to maintenance and reliability problems.

Run-around Coil

As illustrated in Figure 13, a typical run-around coil system is composed of two heat exchangers coupled together by the circulation of

16

an intermediate fluid. The circulating fluid is heated by the hot stream. This heated fluid is then piped to a second heat exchanger whereupon heat is transferred to the process stream. This system could be applied to transfer heat to combustion air, process air, or building air. Since each heat exchanger requires some temperature differential to transfer heat to or from the intermediate fluid, it is inherently less efficient than a direct exchange between the two primary fluids. However, this system is relatively simple and more compact than a direct air/flue gas system. When utilizing the run-around coil for stack gas heat recovery, the heat exchanger in the stack gas stream must be corrosion resistant, but the other heat exchanger can be a compact-finned type. A run-around system circumvents the necessity of close proximity of exhaust and inlet ducts. It is able to transfer heat from one location to another without great retrofit costs.

Direct Contact Heat Exchanger

In a direct-contact heat exchanger, heat is transferred between the two fluids (typically gas and water) without an intervening wall. It is a vertical column in which two fluids move in a counterflow direction, with one of them dispersed as small particles. The direct-contact heat exchanger has received attention due to the fact that there are no heat-transfer surfaces exposed to corrosion, clogging and fouling. Development of this type of heat exchanger has been initiated for desalination plants, cooling towers, condenser units, etc. The absence of tubes or plates reduces maintenance, increases system reliability, and decreases system cost. The elimination of an interfering wall increases the heat transfer rate between two fluids. The direct-contact heat exchanger is an ideal candidate for transferring latent heat from flue gas because a spraying of fine "mist-like" water droplets can provide a large heat transfer surface area in the presence of relatively small temperature differences between the heating and cooling media. Direct contact of a water spray with the flue gas also turns out to be a rather effective low-energy scrubber, which can reduce boiler stack emissions significantly.

17

Conventional Economizer

Preheating feedwater is one of the earliest methods of improving boiler efficiency. An economizer is a device that removes heat from the stack gas to preheat boiler feedwater. The higher feedwater temperature reduces the fuel consumption necessary to generate steam at a fixed rate. To prevent corrosion due to acid condensation, the outer metal temperature of conventional economizers, and hence temperature of the preheated water, is usually maintained above the sulfuric acid dew point of the flue gas (210 to 250 °F). Below this temperature, condensation will occur and corrosion becomes a problem, which reduces the life of the system. Conventional economizers may be fabricated from common heat exchanger materials, and the tubes may be finned to increase the effective heat-transfer area. Finned tubes and common materials promote a compact heat recovery package at a reasonable cost. As mentioned before, a practical rule is that for every 40 °F drop in the flue gas temperature, an increase in the boiler efficiency by 1 perecnt may be expected from the use of an economizer. On the water side, an increase of approximately 1 percent in boiler efficiency may be expected for every 10 °F rise of the feedwater temperature.

Indirect-Contact Condensing Heat Exchanger

The indirect-contact condensing heat exchanger is generally fabricated from corrosion-resistant materials (extruded Teflon, glass, and the like). Since Teflon can only be extruded over smooth surfaces and glass tubes cannot be fabricated as finned tubes, an indirect-contact heat exchanger requires a greater number of tubes and will occupy a greater volume. However, the weight may not be greater since thinner tube walls can be used.

The most popular systems include borosilicate glass tubes, ceramic-coated steel or copper tubes, and teflon-coated copper tubes. Since corrosion is not a factor, the stack gas can be cooled to well below the traditionally recommended minimum safe temperatures (which are about 50 °F above the acid dew point, or around 300 °F) and greater amounts of sensible heat can be recovered. Since much energy within the stack gas exists in the form of latent heat, lowering its temperature below the

water vapor dew point (110 to 140 °F) promotes recovery of large quantities of low grade energy. (This is especially true for fuels such as natural gas, which contain a large amount of hydrogen.) Such a scheme can only be realized using a condensing heat exchanger.

The indirect-contact condensing heat exchanger also acts as a stack gas "scrubber." A report from Brookhaven National Laboratory indicates that lowering the flue gas temperature, by the use of condensing heat exchangers, greatly reduces stack emissions (Ref 3).

Organic Rankine Cycle

The Organic Rankine Cycle (ORC) is one method to convert thermal energy to electrical power. It is a closed loop system filled with an organic fluid having a low boiling temperature. The working fluid is vaporized at an elevated pressure by the waste heat stream in a boiler. This vapor is expanded across a turbine which is directly coupled to an electrical generator. The low pressure vapor from the turbine is condensed in a condenser whereupon it is pumped back to the boiler (see Figure 14). The system is fairly complex with many components. Both the initial investment and annual maintenance can be relatively high.

For a cost-effective application, Ormat Company (Ref 4) (a manufacturer of ORC units) recommends a minimum liquid stream temperature of 230 °F. The same company also suggests that waste-heat streams should exceed 10 MBtu/hr for at least 5,000 hours of operation annually. Most Navy shore facility boilers operate considerably less than 5,000 hours per year with flue gas energy at less than 10 MBtu/hr. Data from various ORC-unit manufacturers have established overall efficiencies of about 10 to 15 percent for waste-heat liquid sources at temperatures of approximately 200 °F. The cost in dollars per kilowatt generated is very high for ORC units rated below 50 kW (Ref 4) and tapers off with increased power ratings above 100 to 200 kW. Operation and maintenance costs for these systems can also be quite high. It is estimated that O&M costs would range between $10,000 and $20,000 per year for a fully loaded unit operating almost continuously.

Thus, both the temperature level and quantity of the waste heat must be sufficient to economically justify the application of ORC. The application of an ORC system would depend heavily on the importance and value of electricity as a commodity. Presently, electricity is priced at more than three times that of fuel oil. Justifying the installation of an ORC system would require a comparison of SIR between alternative means of producing or purchasing electricity.

Heat Pump

Ambient air, ground water, and waste heat streams from industrial processes all contain energy at low temperature levels. Heat pumps have the ability to raise low temperature energy to a higher temperature level. The most common heat pump system is a closed loop filled with refrigerant as shown in Figure 15. A compressor increases the pressure of the refrigerant vapor. In the condenser, the refrigerant vapor is condensed and energy is transferred to process air or water. As the high pressure liquid-refrigerant passes through an expansion valve its pressure is suddenly lowered. In the evaporator, heat is transferred from the low temperature waste-energy source to the colder refrigerant whereupon the refrigerant is vaporized. The refrigerant vapor once again enters the compressor and the cycle continues.

The effectiveness of all mechanically driven vapor compression heat pumps is specified in terms of a Coefficient of Performance (COP) defined as:

$$COP = \frac{\text{Useful thermal energy output}}{\text{Work input to the compressor}}$$

A COP of 5 means using 1 unit of work input (in the form of electrical energy to power the compressor) to deliver 5 units of heat output, of which four units come from the waste-heat source; and all 5 units are now at a raised temperature level.

Although heat pumps are most often driven by electrical motors, heat pumps driven by combustion engines have the advantage of being powered by fuels less expensive than electricity and the added advantage of using the heat given off by the engine. Manufacturers of both gas turbine and

internal combustion engines state that up to 70 perecnt of the heating value of the fuel may be recovered for use in process applications. Driving a compressor with a fuel-fired engine rather than an electric motor becomes attractive when electricity costs are high and natural gas or fuel oil costs are low. However, the initial investment and maintenance costs will be high and overall system reliability will be low.

Because of its ability to raise the temperature level of recovered energy, the heat pump now finds use in industrial plants as a tool for waste-heat management (Ref 5).

BOILER EFFICIENCY IMPROVEMENT

Parameteric Evaluation

Improving boiler efficiency by preheating feedwater and/or combustion air with the stack gas is generally the most effective method of utilizing thermal energy recovered from stack gas (Ref 6). It has been demonstrated (Ref 7) that, compared with using flue-gas heat for process (or building) heating, and for power-generation to drive boiler auxiliaries, feedwater or combustion-air preheating offers the best return in theory. Also, with the feedwater-air preheating scheme, since the recovered energy is available at essentially the same time that it is needed, heat storage equipment (which increases the cost and complexity of the system) may not be necessary. Realistic economic justification is largely dependent on whether the boiler can fully utilize the recovered energy. For example, the temperature of boiler condensate return typically ranges from 140 to 180 °F, while that of makeup water is approximately 60 to 80 °F. A steam system which returns 100 percent condensate will then have limited options for improving boiler efficiency, while a "once-through" system (which requires 100 percent makeup water) has excellent potential to use recovered energy and improve boiler efficiency. In many cases, boilers may operate with low quantities of makeup water. In such cases, condensing heat exchangers offer little economic benefit if the sole purpose is to improve boiler efficiency by preheating makeup water. Due

21

to the superior heat transfer qualities of water, preheating feed water is more efficient and economical than preheating combustion air. However, air is always available at ambient temperature with good potential for accepting flue gas heat.

In any application, usually more than one kind of heat recovery system can be utilized to improve boiler efficiency. In order to determine the most promising alternatives, a parametric evaluation technique was used in the present study. Relevant parameters affecting a heat recovery system were identified, and each parameter was given a weighting factor based upon its overall importance. Various heat recovery schemes were then judged on each of these parameters, using a scale of 1 for unsatisfactory to 10 for outstanding. An overall score for each heat recovery scheme was then obtained by summing the products of the weighting factors and the respective parameter score. Table 3 shows the result of this computation.

The indirect-contact condensing heat exchanger and the conventional economizer systems were found to be the two most favorable systems. The conventional economizer was most cost effective, above average reliability, lower O&M costs, and is compact in size. The indirect-contact condensing heat exchanger has a favorable cost effectiveness along with potentially high energy savings. Both the conventional economizer and the indirect-contact condensing heat exchanger can be utilized to improve boiler efficiency by preheating feedwater. Air preheating systems which were favorably ranked include a run-around coil system and an air-to-air heat exchanger.

Based on the above assessment, the conventional economizer and the indirect-contact condensing heat exchanger are discussed extensively in what follows. The combustion-air preheaters are then described briefly. Finally, the direct-contact heat exchangers are covered in detail under the heading PROCESS-AIR-WATER PREHEAT.

Conventional Economizer

System Operation. Boiler efficiency improvement with a conventional economizer is dependent upon the type of fuel burned and the stack gas temperature as shown in Figure 16. It is, however, independent of the

22

quantity of makeup water. Boiler condensate return and makeup feedwater are both heated first in the deaerator. This warm feedwater is then fed to the economizer where it is further heated by the stack gas. The temperature of the stack gas leaving the economizer and that of the feedwater entering the economizer are limited to certain levels to protect equipment from corrosive conditions. The process flow diagram in Figure 17 illustrates a boiler plant equipped with an economizer to recover waste heat in the stack gas.

Corrosion. The major portion of sulfur in fuel is burned and appears as sulfur dioxide in the stack gas; a small portion (3 to 5 percent) is further oxidized to sulfur trioxide. These oxides combine with the moisture in the flue gas to form sulfurous and sulfuric acid vapors. When in contact with a surface below the acid dew point, condensation takes place. The acid dew point is directly related to the amount of sulfur in the fuel as shown in Figure 18.

Because of the higher heat transfer coefficient in liquid-metal heat transfer than in gas-metal heat transfer, the gas side metal temperature of an economizer is closer to the water temperature than to the gas temperature. Therefore, to avoid condensation and subsequent corrosion problems, the economizer feedwater temperature must be, on the average, no lower than 20 to 50 °F below the acid dewpoint. Figure 19 illustrates the minimum recommended feedwater temperature necessary to avoid sulfuric acid condensation at varying fuel sulfur concentrations. There are a number of methods available to reduce or eliminate this "cold-end" corrosion problem (Ref 8). They are briefly described below:

1. **Reduce Excess Air.** Reducing the excess air decreases the quantity of sulfuric acid vapor within the stack gas. Research indicates a direct relationship between sulfur trioxide formation and excess oxygen (or air) levels (Ref 7).

2. **Use Corrosion-Resistant Metals or Sleeves.** Corrosion-resistant alloy steels can be used in preventing corrosion; however, their high cost normally prohibits their actual use. Interlocking cast iron sleeves over carbon steel tubes can also be used where severe acid conditions are

23

anticipated. The cast iron sleeve will corrode, but it can be replaced as necessary at normal maintenance intervals. The cast iron sleeves protect the carbon steel tubes from contact with the corrosive gases.

3. **Improve Fluid Flow Arrangement**. Using a parallel-flow tube arrangement rather than a counterflow arrangement increases economizer skin temperature. With a parallel flow arrangement, cold feedwater enters where the stack gas is hot; thus, the coldest spot on the outside tube surface will be hotter.

4. **Heat Feedwater Before Entering Economizer**. The feedwater can be heated well above the minimum metal temperature to assure that corrosion does not occur. However, the heat recovery capability is then reduced for two reasons. First, the feedwater has less capacity now to pick up heat; and secondly, it takes added energy to heat the feedwater to these higher temperature levels before entering the economizer. Feedwater heating can generally be accomplished with a deaerator before the economizer. Most boilers are already equipped with deaerators to protect the tubes from internal corrosion caused by the dissolved oxygen in the feedwater. Consequently, an existing deaerator can serve a dual purpose and protect both boiler tubes from internal corrosion and the economizer tubes from external corrosion.

5. **Modulate Feedwater Flow Through the Economizer**. One may vary the feedwater flow rate through the economizer so that minimum metal temperatures are kept above the acid dew point by diverting feedwater flow around the economizer during periods of low flue gas temperatures.

Maintenance. For an economizer, maintenance primarily consists of keeping heat transfer surfaces clean. Both soot and slag deposits reduce heat transfer and increase corrosion. Maintenance is commonly performed through steam or air blowing, or water washing.

Sootblowers are normally included as part of an economizer system. They blow on the economizer tubes at programmed time intervals or on a manual basis, depending on the type of tube deposits and fuel burned. For example, sootblowing may be necessary only once a day for a gas-fired system; while once an hour is more common for oil- or coal-fired systems.

Materials. Cast iron is the most common material used for economizers because it is economical and exhibits acceptable acid corrosion resistance. Figure 20 shows a comparison of relative corrosion rates among some common materials in low temperature flue gas.

Minimum Flue Gas Temperature Leaving Economizer. To avoid corrosion downstream of the economizer, the stack gas exit temperature should be well above the acid dew point. Minimum recommended temperatures are based on the sulfur content of the fuel. Recommended values are: 300 °F when using natural gas as fuel; 325 °F for No. 2 fuel oil; and 350 °F when using No. 5 fuel oil, No. 6 fuel oil, or coal.

Controls. Three basic control schemes exist to maintain reasonable economizer skin temperature and stack gas exit temperature to avoid economizer and downstream stack corrosion. They are summarized as below.

1. Below a certain exit temperature, stack gas is directed around the economizer. This scheme protects the stack and fan downstream of the economizer; but it relies on adequate feedwater preheat across the deaerator to protect the economizer. This procedure require installing the economizer in parallel with the existing stack, rather than the more common "in-line."

2. The feedwater flowrate into the economizer is regulated to maintain the gas exit temperature above the minimum recommended level to avoid downstream stack and fan corrosion. A three-way modulating valve controls the amount of feedwater entering the economizer, and the rest of the feedwater bypasses the economizer as illustrated in Figure 21.

This bypass scheme strives to maintain a safe stack gas exit temperature. The deaerator is again depended upon to maintain minimum metal temperatures, and to avoid tube corrosion.

3. A scheme that controls the feedwater temperature entering the economizer is shown in Figure 22. An auxiliary steam heat exchanger is installed between the deaerator and the economizer. Temperature sensors monitor the stack gas temperature leaving the economizer and the water temperature entering the economizer. These temperatures are input to a controller which regulates the steam flow into the auxiliary heat exchanger, thus maintaining both the economizer exit gas temperature and inlet feedwater temperature above the corrosion-free levels.

In most cases, the vendor will propose a control scheme as part of the economizer "package." Such control systems should not be considered accessories but rather a necessary part of the system.

Retrofit. A simplified economizer retrofit arrangement with temperature design limitations is shown in Figure 23. Although a parallel-flow exchanger provides higher metal surface temperatures needed for good corrosion control, the more effective counterflow heat exchanger with a good control system is frequently suggested by vendors. Upward flow of feedwater is a benefit because it avoids air buildup within the system. The maximum feedwater preheat temperature should be 35 to 75 °F below the saturation temperature of the boiler to prevent steaming, water hammer, and thermal shock.

For a clean fuel such as natural gas, a staggered-tube arrangement may be used. For other fuels, an in-line arrangement is necessary to combat tubing deposit buildups and to avoid plugging.

Soot blowers or a water wash system should be installed above the economizer and at points along the depth of the economizer to assure thorough cleaning of the heat transfer surfaces. Below the economizer, a hopper with a flanged valve would be needed as a hookup to a waste removal system.

Most packaged watertube and firetube boilers use the forced-draft system. With such a system, an economizer can generally be installed without an additional fan. When boiler systems are designed, fans are traditionally oversized by about 10 to 15 percent, both in volume, flow rate, and in pressure rise. This fact, along with a reduced firing rate (and a consequently reduced combustion air flow rate), in most cases, compensates for the pressure drop across the economizer. Some economizers are intentionally designed for gas side pressure drops of less than 1.0 inch WC to avoid the need for an additional fan. The only exception is the natural draft boiler where the reduced stack gas temperature decreases the needed draft through the boiler. This reduction in boiler draft and the added pressure drop due to the heat recovery equipment would make the purchase of an additional fan necessary in the case of a natural-draft system.

Load Used in Sizing. Boiler load varies throughout the year. If an economizer is designed for the maximum rated load, it will be a large and expensive system. If low load is used in sizing, the economizer will be less effective at higher operating loads and will produce a larger pressure drop. When retrofitting within a compact space, sizing based on the average yearly load may be necessary. Figure 24 illustrates what can be expected when operating an economizer at off-design condition, without controls. At a mass flux larger than the designed value, a larger amount of heat can be recovered but at a higher pressure drop. At a lower mass flux, the rate of heat recovery is reduced and the stack gas exit temperature is lowered toward the acid dewpoint, where corrosion starts.

Miscellaneous. Some boilers may already be equipped with conventional economizers. Other heat recovery systems should then be considered to further capture the stack gas energy.

Installed cost of an economizer system is difficult to estimate. The cost of hardware may only be a small fraction of the total installed cost, and the economizer itself may only represent a fraction of the total hardware cost. A vendor, in many cases, will quote the hardware cost of the economizer alone. A buyer should then be aware that a complete set

27

of controls is necessary, and may cost as much as the economizer itself for smaller systems. Design, fabrication, and installation of structural support are other costs to be included in the system budget. Overall economics will be highly dependent upon the physical arrangement of the boiler, breaching, and stack.

On an average, a conventional economizer can improve boiler efficiency by about 1.5 to 3.5 percent with a SIR of between 4 and 12. Although the efficiency improvement is small, the installed cost of an economizer is significantly lower than the cost of a condensing heat recovery systems. An economizer system is recommended where the average stack gas temperature exceeds 450 °F, the annual boiler operating time is longer than 2,500 hours, and the stack gas flow rate is greater than 15,000 lbm/hr. The greatest advantage of an economizer is probably its compact size and low cost. Figure 25 provides a rough idea of its relative size when compared with direct contact heat exchangers and indirect contact condensing heat exchangers.

Design Considerations. The first step in designing an economizer is to determine the mass flow rates of the flue gas, water, and fuel. The economizer is usually modeled as a rectangular-shell, in-line, finned-tube heat exchanger fabricated from cast iron. Typical design parameters for selected fuels are given in Table 4. Fins serve to increase the heat-transfer surface area. Thick fins offer greater erosion protection and greater heat transfer effectiveness. Thin and tall fins offer less erosion resistance and are less effective, but exhibit greater heat transfer area for less cost. Fin spacing is chosen according to the type of fuel burned: for clean-firing fuels, 54 to 60 fins/ft; for distillate fuels, 48 fins/ft; for heavy oil, 36 fins/ft; and for coal, 24 fins/ft. An economizer with closer fin spacing is more compact and cost effective. High-fouling fuels require wide fin spacing to avoid soot accumulation, fouling, and clogging. Typical fin heights for tube diameters from 1.5 to 2.0 inches range from 0.75 to 0.875 inch with thicknesses ranging from 0.060 to 0.075 inch. For coal applications, a minimum fin thickness of 0.105 inch is recommended to cope with erosion.

In practice, a boiler will operate at a variety of load conditions throughout the year. Instead of optimizing the economizer based on average operating conditions, a more exact breakdown of the boiler operating conditions will provide a closer optimization. The total annual energy savings can then be estimated by a summation of energy savings at each load condition.

The detailed design procedure for an economizer is routinely available in standard references. A design software based on the procedure and data presented in Reference 9 is also commercially available for IBM-PC and compatibles: COMPACT HEAT EXCHANGER DESIGN, Intercept Software (3425.S Bascom Ave., Campbell, CA 95008). For specific cases encountered in practice, NCEL is ready to offer its services in a consulting capacity; please contact Dr. T.T. Fu at NCEL.

Indirect-Contact Condensing Heat Exchanger

System Operation. Stack gas heat recovery in the condensing mode consists of two basic stages. In the first stage, mostly sensible heat and a small portion of the latent heat is removed. In the second stage, the stack gas is lowered below the water vapor dewpoint thereby recovering most of the remaining latent heat.

A boiler requiring a large amount of makeup water is a prime candidate for a condensing heat recovery system because large amounts of heat can be transferred effectively from the stack gas to the cold makeup water.

On a system level, one of two options is possible. In one case, only the makeup water is preheated, pumped to the condensate storage tank, and is further heated by the deaerator before entering the boiler (see Figure 26). Due to low temperature of the makeup water, thermal energy recovery from the stack gas can condense about 40 percent of the water vapor from the stack gas. In the second case, both makeup water and condensate return are mixed in the condensate storage tank before being preheated in the indirect-contact condensing heat exchanger. The boiler feedwater is then pumped to the deaerator where it is further heated, if necessary, before entering the boiler (see Figure 27). With this scheme, however, the latent heat recovery is rather limited.

Materials. Since the indirect-contact condensing heat recovery system is subjected to a corrosive environment, it is important that materials employed be corrosion resistant, and have a reasonably long operating life. They must be capable of withstanding sulfuric and sulfurous acids. The typical corrosion zones of stack gas are shown in Figure 28 and the corrosion rates of various materials in terms of sulfuric acid concentration and temperature are shown in Figure 29.

Two types of corrosion-resistant heat exchangers are commercially available: One fabricated from borosilicate glass tubes, and the other from Teflon-coated copper tubes. Although both materials do not conduct heat well, overall heat transfer is not significantly affected since gas-side thermal resistance dominates the heat transfer. Both materials have low friction coefficients, which reduce fouling and increase maintainability. The smooth surfaces enhance heat recovery via the more efficient dropwise condensation rather than filmwise condensation.

Glass tube heat exchangers are limited to applications where the flue gas temperature does not exceed 400 °F, and the water pressure does not exceed 50 psig. Therefore, the saturation temperature of water at 50 psig of 298 °F becomes the maximum design temperature. A maximum operating feedwater preheat temperature of 250 to 275 °F is suggested.

Teflon can be extruded over tubing as a thin film (0.015 inch). It can operate with flue gas temperatures up to 500 °F (400 °F recommended for continuous operation) and a maximum design water pressure of 150 psig. Teflon-coated heat exchangers are capable of raising water temperature to 200 to 250 °F.

Another less common type of heat exchanger uses stainless steel tubes. The American Medical Institute of Los Angeles, Calif., has installed stainless steel heat exchangers at 25 hospitals with gas-fired boilers. Some of these heat exchangers have been operating trouble-free for the last 8 years. Stainless steel 304 has been shown to be a durable material in a stack gas environment from a gas-fired boiler if chloride levels are not unusually high (Ref 10). Stainless Steel Type 304, along with other austenitic stainless steels should not be used where there is potential chloride contamination as chloride stress corrosion cracking may be encountered at temperatures in excess of 100 °F.

For stacks and ducting, fiberglass-reinforced plastic (FRP) and acid resistant brick lining are the two most popular materials used to prevent corrosion. FRP can be used up to 300 °F, but 160 °F is probably a conservative maximum to be observed for long-term operations because temperature excursions can be detrimental to FRP systems. At higher temperatures (up to 850 °F), foamed borosilicate glass can be used as a lining to protect the FRP from high temperatures. The growing use of ceramic/refractory materials, such as borosilicate glass, speaks for their low cost and higher abrasion resistance as compared to alloy steels. Another commercially available product is porcelain enamels (glass linings). Porcelian enamels can be bonded to steel by fusing a glass or ceramic frit on the steel surface at high temperatu. The bond, however, is <u>not</u> inseparable. Compared to brick-lined stacks, glass-lined stacks are considerably lighter and require less foundation and support. This technology is still being developed and improved.

With an indirect-contact condensing heat recovery system, a fan is needed to overcome the pressure drop across the heat exchanger. If an induced-draft fan is installed downstream of the heat exchanger, it must be constructed of either fiberglass or stainless steel. If the fan is placed upstream of the heat exchanger, the fan does not have to be fabricated from FRP, but it must be suitable for high temperature applications.

Flue gas desulfurization (FGD) units are becoming more widely used. Adding a FGD scrubber to a boiler system typically reduces the gas temperature to a range where acid condensation occurs. Experience with scrubber materials has greatly enhanced the knowledge of materials operating under stack-gas acid conditions. A summary of actual applications of FRP equipment operating in corrosive environments can be found in Table 5.

Figure 30 shows key components, and respective materials used within an indirect-contact condensing heat recovery system.

Controls. While the control strategy for the conventional economizer is based on maintaining metal temperatures above the acid dew-point to mitigate corrosion, the control of an indirect-contact condensing system aims at protecting materials against excessive temperatures.

A typical control scheme consists of monitoring the stack gas temperatures entering and leaving the heat exchanger. If the safe operating temperature of the material is exceeded, the controls will either stop flow to the heat recovery system and shut down the fan, or modulate an ambient air damper to temper the stack gas temperature.

Controls also act to shut down the heat recovery system if the heat exchanger water-exit temperature exceeds a set point, or upon loss of water flow through the heat exchanger.

Maintenance. Maintenance of an indirect-contact condensing heat exchanger involves conditioning and dispensing of the condensate. Because the condensate will be somewhat acidic, 2 to 4 pH, it will have to be neutralized before disposal. Filters may also be needed to remove entrained solids. However, experience indicates that the stack gas condensate can be reasonably clean, and requires minimal conditioning (Ref 11). A gas-fired boiler would require a smaller conditioning and disposal system than an oil-fired one, due to lower levels of sulfur dioxide and particulate matter in the stack gas.

Due to the excellent fouling resistance of both the Teflon-coated heat exchanger and the glass heat exchanger, a simple intermittent water spray system is all that is needed to maintain efficient system performance.

Miscellaneous. Indirect-contact condensing heat exchangers are generally sold as packaged units. Pumps, fans, controllers, structural framework, etc. are all preselected and included in the total price. Beyond the package cost, there are additional costs including retrofitting ductwork and piping. This system can be operated with gas- or oil-fired boilers. The condensing mode of operation recommends itself to boilers using natural gas because of the sizable amount of water vapor in the flue gas.

The heat exchanger system is installed near the existing boiler stack to reduce installation costs and to permit easy isolation of the heat exchanger without having to shut down the boiler plant. Isolation of the heat exchanger is necessary under the following circumstances:

1. If the process dependent upon the waste heat is shut down, the stack gas can be easily diverted away from the heat exchanger.

2. If the heat exchanger itself is not operating, or needs maintenance, the stack gas can easily bypass the heat exchanger.

3. If the stack gas temperature or flowrate is high, a portion of the stack gas can be diverted away from the heat exchanger, assuring that materials within the system are protected and water preheat is not excessive.

The greatest advantage of a corrosion-resistant heat exchanger is its ability to recover stack gas energy below the acid dew point and not necessarily below the water vapor dew point. Many currently operating higher efficiency systems still do not reach the condensing stage; they succeed only in removing most of the sensible heat from the stack gas. Lowering the stack gas temperature below the water vapor dew point to recover large quantities of low temperature energy should be carried out only when this energy can be used in a cost effective manner.

Indirect-contact condensing heat exchangers can significantly improve boiler efficiency. The actual improvement is largely determined by the stack gas temperature and the makeup feedwater fraction of the total feedwater. As shown in Figure 31, even with a stack gas temperature as low as 300 °F, boiler efficiency can be improved by as much as 7 percent, where conventional economizers could not even be used.

Combustion-Air Preheaters

Many air-preheating systems are commercially available. The great majority of such systems are fabricated from common materials and will not last long in a corrosive environment. These heat exchangers are generally installed "in-line" with the existing stack, with minimal precautions against "cold-end" corrosion. Since air and stack gas generally have about the same heat transfer characteristics, the metal surface temperature can be roughly approximated as the average of the inlet stack gas and air temperatures if the two flowstreams pass in a parallel fashion. This metal temperature can then be compared with the minimum metal temperature necessary to avoid corrosion.

33

At low boiler loads, the stack gas temperature tends to be lower and corrosion can be a problem, unless the flue gas is diverted around the heat exchanger. Corrosion-resistant air/stack-gas heat exchangers are commercially available both in the Teflon-coated plate type and the glass tube type. Both types of heat exchangers can reduce stack gas temperature below the sulfuric acid dewpoint with no danger of corrosive action.

Heat exchangers for air/stack-gas systems tend to be rather large due to poor gas to metal heat transfer rates. Such systems also require ducting of the preheated air to the boiler inlet, taking up valuable space within boiler rooms. In general, retrofit installations cost 4 to 8 times that of the system package itself.

A run-around coil system consists of two liquid/air heat exchangers, piping, and a pump. One heat exchanger is installed in the path of the stack gas and the other in the combustion airstream. It is a more compact and flexible setup than a direct air/stack-gas system. Since heat can be inexpensively transported from one location to another with a circulating fluid, the stack gas and the combustion air streams do not have to be in close proximity. Furthermore, only the heat exchanger in the pathway of the stack gas needs to be corrosion resistant. Run-around coil systems are common with HVAC energy conservation applications such as preheating building makeup air. These systems are not prepackaged, and some vendors have no experience at all with the handling of stack gas streams.

A hybrid heat recovery system, preheating both makeup feedwater and combustion air, can be an excellent method for recovering a large amount of the energy from the stack gas to improve boiler efficiency. If a hybrid system is used in modular form, some modules could be used to preheat feedwater and others, as part of a run-around coil, to heat combustion air. The hybrid heat recovery system would be, of course, more complex and expensive.

Most packaged boilers are equipped with forced-draft fans, which are an integral part of the boiler. The addition of a combustion air-preheat system would, therefore, require retrofit work at the location of the forced-draft fan. The existing fan would have to be moved away from the boiler and the heat exchanger installed between it and the boiler.

34

Fan horsepower would have to be adequate to overcome the pressure drop of
the additional ducting and heat exchanger. If the preheated combustion
air exceeds a rated temperature, additional hardware change may be needed.
The first step in considering combustion-air preheat would be, thus, to
consult the boiler manufacturer. In general, firetube boilers are not
designed to operate with preheated combustion air, and it would be imprac-
tical to attempt improving efficiency in these boiler systems.

PROCESS-AIR-WATER PREHEAT

Applications

As stated earlier, improving boiler efficiency is generally the most
attractive method of using stack gas heat. However, in many instances,
it may not be possible to economically justify the installation of heat
recovery equipment to improve boiler efficiency, because of the relatively
small amount of energy which can be fed back to the boiler. In such cases,
alternative methods must be considered. Such alternatives include heating
buildings, preheating water for baths, kitchens, laundrys, industrial
processes, etc.

Many successful stack gas heat recovery applications have involved
preheating process fluids. The explanation for this success is simple.
Many processes require large quantities of warm water on a continuous
basis throughout the year. A lower process temperature directly trans-
lates to large amounts of low-quality stack gas energy that can be easily
recovered and utilized. Since greater amounts of energy are recovered,
the annual energy savings and the life cycle savings are greater. Since
the energy recovered need not be at high temperature levels, the actual
recovery itself can be easy and cheap. A disadvantage of process preheat
applications is that they often require storage tanks, series of heat
exchangers, more extensive controls, etc., which may erode the overall
cost effectiveness.

Building Preheat. One advantage of heating a building is that the
heating load is in phase with the boiler-operation load. For example,
in cold northern states, heating is needed continuously during the winter
season; but during the late fall, winter, and early spring, the boiler
is also operating nearly full-time. As mentioned before, the operating
schedule is an important factor when justifying a heat recovery installa-
tion. This in-phase matching then makes the scheme inherently attractive.
The major disadvantage here is the need for an extensive piping or ducting
system in the building. Unless the existing piping or ducting network
can be utilized, retrofit for heat distribution could be expensive.

Industrial Processes. Many industrial processes require boilers
which operate a great number of hours annually. Such processes frequently
have a need for water at varying temperature levels. In such instances,
heat recovery systems have been implemented successfully in applications
ranging from providing hot water for washing cans in tomato factories to
using the water in textile drying and finishing processes. Payback periods
for these heat recovery installations can be very short.

Hot Water for Laundries, Kitchens, Baths, etc. Hot water is always
needed for washing dishes, taking baths, washing clothes, etc. Boilers
which are physically located close to such activities represent heat
recovery possibilities. Storage tanks are normally necessary since the
load varies throughout a day.

Several types of heat exchangers can be employed to preheat process
water or air. As carried out for boiler efficiency improvement, these
candidate schemes can be evaluated parametrically, as presented Table 6.
Both the direct-contact type and the indirect-contact type of condensing
heat exchangers rank very high, having short payback periods with large
energy savings. The following discussion addresses only the direct-
contact heat exchanger.

Direct-Contact Heat Exchanger

System Operation. A direct-contact heat exchanger has a set of
nozzles positioned at the top of a column, spraying water downward, in
counterflow with the stack gas. The bottom of the column acts as a res-
ervoir. A pump recirculates this warm spray water from the reservoir at
the bottom, through filters and an intermediate heat exchanger, to the
spray nozzles. A secondary circuit pumps the process fluid through the
intermediate heat exchanger where it is heated (see Figure 32). Because
the circulating spray water can only attain a temperature equivalent to
the adiabatic saturation temperature of the stack gas (typically a maxi-
mum of 130 °F), heat can only be transferred to fluids at temperatures
lower than 130 °F. With this system, there is no contamination of the
process fluid. However, the circulating spray water must be continu-
ously filtered to remove particulate matter scrubbed from the stack gas,
and chemically neutralized and drained on a regular basis to minimize
corrosion of the intermediate heat exchanger and piping system.

A direct-contact heat exchanger system may be considered to consist
of two stages (Figure 32):

1. The humidification stage, where stack gas enters the heat ex-
changer and is humidified by the water spray. Sensible heat is removed
from the stack gas as it is cooled to its adiabatic saturation tempera-
ture.

2. The dehumidification stage, where water is condensed from the
stack gas thereby removing a portion of the latent heat from the stack
gas.

Thermodynamic Limitation. The adiabatic saturation temperature
represents the thermodynamic limitation on the hot water temperature
attainable through a direct-contact heat exchange process. It is depen-
dent upon the inlet gas temperature and humidity, as shown in Figure 33.
A distinction must be made in principle between the stack gas dew point
as it exits the boiler and the adiabatic saturation temperature of the
stack gas as it leaves the humidification stage of the heat exchanger.
As stack gas exits the boiler, the dew point is solely dependent on the

amount of excess air and the type of fuel fired by the boiler. As the
stack gas leaves the heat exchanger's humidification stage, it will con-
tain a greater amount of water vapor, close to saturation at the prevail-
ing temperature.

Materials. The spray plenum is generally formed from an insulated,
stainless steel material. Fiberglass-reinforced plastic (FRP) cannot be
used here because temperatures along the plenum can be quite high. In
the case of a gas-fired boiler, the primary pump and the secondary heat
exchanger can be fabricated from 316 stainless steel. In the case of an
oil-fired boiler, 316 stainless steel may be adequate but the intermedi-
ate heat exchanger would require a higher nickel content alloy, such as
Hastelloy G, or a corrosion-resistant material, such as FRP or Teflon.
The intermediate heat exchanger is normally of the plate type.

Heat Transfer Augmentation. Figure 34 illustrates two common ways
to augment the spray mode of heat transfer. The baffle-plate unit in-
creases heat transfer by directing the stack gas around the plates to
increase its residence time in the heat exchanger. Alternatively, a
packed tower contains materials of various geometric shapes designed to
increase heat transfer surface area. A compromise is made between en-
hanced heat transfer and increased fan power due to a larger gas pres-
sure drop (Table 7). Another novel way to enhance heat exchange is to
raise the limitation of the adiabatic saturation temperature by spraying
liquids other than water. More research and development work is still
needed in this direction.

Types of Fuel. Although the principal advantage of a direct-contact
heat recovery system is its ability to deal with "dirty" and corrosive
stack gas, most case studies involve relatively clean natural gas-fired
boilers. Condensing heat recovery schemes are ideal for gas-fired boil-
ers because a large portion of the recoverable energy in the stack gas
is in the form of water vapor. Direct-contact heat exchangers can be
used with a light fuel oil; but, as the ash and sulfur content of the
fuel increase, corrosion problems increase. Coal is not currently used
with this kind of system because of severe corrosion/erosion problems

38

associated with ash, sulfur, and other contaminants. In addition, coal produces inherently smaller amount of water vapor and is also relatively low-priced, thereby reducing the economic benefits of this approach.

Control System. The control system for a direct-contact heat exchanger is fairly simple. The circulating water spray is heated by the stack gas and ends up in the reservoir at the base. The flow rate of the spray water can be controlled based on a minimum low-level feedwater temperature, or on a certain fixed stack gas exit temperature. A variable speed fan or a modulating damper could be used to provide a constant stack gas pressure at the boiler outlet. In the event that makeup water is temporarily not needed, the spray water will stabilize at its adiabatic saturation temperature and the unit will simply act as a scrubber. If there is no demand for heat from the circulating spray water, stack gas can be diverted to the original stack with bypass dampers and the circulating pump shutdown. Like the indirect-contact condensing heat exchanger, this system must also be equipped with a pH control system. A chemical metering pump is needed to maintain the pH of the circulating spray water at acceptable levels, or the spray system must be continually diluted to acceptable levels. Because water is constantly being condensed from the stack gas, an overflow system is needed to dispose the additional water.

Maintenance. Corrosion due to acidic water and waste disposal can be a significant problem. Continuous chemical treatment of the spray water or dilution by freshwater is needed to maintain a pH above 2. The spray water must be continuously filtered and cleaned to remove particulate matter that could damage pumps, block lines, and interfere with an efficient spray system. A direct-contact heat exchanger does not require any additional maintenance efforts other than those ordinarily associated with pumps, fans, dampers, etc.

Miscellaneous. When operating with a gas-fired boiler, the direct-contact system has been proven reliable and maintenance free (Ref 12). For oil- and coal-fired systems, maintenance costs will be higher and reliability will be lower than for gas-fired boilers.

More than 700 such systems are currently in operation in Europe, some successfully for 6 to 8 years (Ref 12). Most of these systems are used in gas-fired boilers to heat large buildings or swimming pools. Figure 35 illustrates a direct-contact heat exchanger linked with a run-around coil to preheat building makeup air. An added advantage of this kind of heat exchanger is its excellent pollution control. A direct-contact heat exchanger essentially acts as a low-energy scrubber. Particulate collection efficiency can be as high as 90 to 95 percent (Ref 13).

When 100 percent makeup water is involved, a boiler efficiency improvement of at most 5 percent can be expected if preheated by a direct contact heat exchanger. For this reason, these heat recovery systems are mainly utilized to provide heat to low temperature process operations. A direct-contact heat exchanger can save energy in the range of 10 to 15 percent with a SIR of over 6, if (1) the boiler is fired by either natural gas or light fuel oil (distillate), (2) the boiler is operated for at least 4,000 hours annually, and (3) at least an energy amounting to 10 to 15 percent of the boiler fuel input is needed for low temperature processes.

Parameters affecting direct-contact heat exchanger design can be found in Reference 14.

INDUSTRIAL EXPERIENCE IN WASTE HEAT RECOVERY

Studies of existing cases are important in the field of waste heat recovery (Ref 15). Some actual cases have been collected from a variety of sources as general references, and are included here for the benefit of the reader.

Case 1

A hospital has a 15 MBtu/hr gas-fired boiler operating at an average load of 10 MBtu/hr (which reduced to 5 MBtu/hr in the summer). The combustion is at 3 percent excess oxygen, and the flue gas temperature is 450 °F. The original boiler efficiency was 75.5 percent. The dry flue-gas flow rate was 12,000 lbm/hr. With an estimated humidity of 0.11 lbm/lbm dry gas, the heat recoverable by feedwater through a conventional

economizer was estimated to be 1 MBtu/hr (with an equivalent new boiler efficiency of 83.4 percent). If an indirect-contact condensing heat exchanger were used, the heat recovered would be 1.3M Btu/hr with an improved boiler efficiency of 85.5 percent. The plan with the conventional economizer was eventually adopted in view of its simplicity and shorter payback period.

Case 2

The natural-gas-fired boiler at the Royal Ordnance Factory at Chorley, England, was retrofitted with a Dumista spray recuperator manufactured by Bayliss Kenton Installation, Ltd. of Blackburn, England, and the boiler efficiency has been reported to be improved by 12 percent. The heated water from the recuperator is pumped through a plate heat exchanger which transfers heat from the primary circuit to the cold makeup feedwater.

Case 3

In a paper mill, the exhaust gas from the paper dryer was a source of waste heat estimated at 30 million Btu/hr, 70 percent of which was latent heat. The burner was already equipped with a combustion air preheater using a plate heat exchanger which recovered 2 million Btu/hr, the remaining 28 million Btu/hr was wasted. A preliminary decision regarding further recovery centered on gas/water heat exchange options: a waste-heat boiler, economizer, and direct-contact spray recuperator. The low temperature level excluded the waste-heat boiler from further consideration. An engineering study showed that an economizer could recover 8 million Btu/hr of heat, whereas a spray recuperator could recover about 17 million Btu/hr. The fact that the exhaust gas was loaded with fiber made the spray recuperator a prime candidate since it could also wash the exhaust stream, reducing the emission of solid pollutants. The spray water was isolated from the clean water by employing a liquid/liquid plate heat exchanger.

The unit heats up clean water to 104 °F, which was usable elsewhere in the plant. The resulting clean water supply accounts for about 14 million Btu/hr recovered from the exhaust-gas.

Case 4

An exhaust stream carried about 12 million Btu/hr of waste heat (20 percent sensible and 80 percent latent). The temperature level of the stream was 415 °F. It was planned to utilize 2 million Btu/hr of the sensible heat by preheating air from 81 °F to 320 °F. A single, finned heat pipe was tested in the exhaust flue, with the condenser end of the pipe cooled to simulate expected operating conditions. The test showed no perceptible corrosion due to acid condensation. There was, however, a fouling problem due to condensation of wax carried from the plant. This condensed wax could only be removed by wire brushing and high pressure water jet. Thus, devices like the plate heat exchanger and the heat wheel could not be adopted, since they are easily fouled and difficult to clean. The decision was then made to use heat pipes with individual banks removable for cleaning at regular intervals.

Case 5

The stack gas from a malt kiln varied from 86 °F at 95 percent humidity to 185 °F at 5 percent humidity for different phases of the malting operation. About 25 percent reduction in total energy consumption was realized for several years by using a copper run-around coil with aluminum finned heat exchangers to preheat the combustion air, the air inlet being remote from the kiln exhaust. But a recent change in the process operation gave rise to acid exhaust conditions; and within a short time, the heat exchanger at the exhaust side corroded and virtually disintegrated. Things are back to normal again after replacing the exhaust-side heat exchanger with one having glass tubes.

Case 6

The concern about the reliability of a glass tube heat exchanger in the presence of daily mechanical disturbances may be lessened somewhat by the following installation, which was completed in 1978 and has been in satisfactory operation ever since. A brewery has a malt bed through which air is passed. The exhaust air is warm and moist. A glass tube heat exchanger recovers sensible and latent heat from the exhaust air to preheat the supply air. The process is in operation 20 hours per day and 360 days per year. The exchanger was made of about 10,000 borosilicate glass tubes. The supply air is heated from 50 °F to 75 to 127 °F. The final temperature depends on moisture level in the exhaust air. It has been estimated that the saving on gas-fuel is about $20,000/yr, from which an amount of $1,500/yr is subtracted to account for the additional fan power. The installed cost of the exchanger of $50,000 was recovered in about 3 years. After years of operation, only the mild steel frame showed any sign of corrosion, which could have been avoided if stainless steel were used in the design. The free flowing condensate along the vertical tube surfaces apparently washed away fouling particulates before they settled. The unit is still operating and its effectiveness is reported to be between 70 and 77 percent.

Case 7

A stainless steel condensing heat exchanger has been recovering waste heat from an 8.50-MBtu/hr natural-gas-fired boiler since 1980. The largest problem encountered has been damper linkages binding up. Maintenance has been minimal. Flue gas temperature is lowered from 350 °F to 150 °F. Water is preheated from 100 °F to 120 °F and utilized for hospital laundry service. A 1,000-gallon tank stores hot water when the laundry service is not operating. Fuel savings are approximately 5 percent.

Case 8

A linen service company has successfully implemented a Teflon-coated copper heat exchanger to preheat process water from 50 °F to 90 to 100 °F. Boiler flue gas temperature is lowered from 450 to 500 °F to 120 °F. Fuel savings range from 8 to 10 percent. The boiler is fired by both No. 6 fuel oil and natural gas. The unit has been in operation since 1983 with no significant maintenance problems.

CONCLUSIONS

The largest energy loss of a boiler is due to hot flue gas leaving through the stack. On the average, this loss accounts for approximately 17 percent of the fuel input. With a heat recovery system, it is possible to recover up to 60 percent of the stack losses. It has been shown that this recovered energy can be more effectively utilized by preheating boiler feedwater or combustion air than using it for process heating. However, practical considerations may determine process heating as the most cost effective application. Due to the superior thermal properties of water, heating feedwater is more effective than heating air. Research in heat-recovery material technology has identified materials which are able to deal with sulfuric acid condensation occurring in flue-gas condensing heat-recovery systems. Some of these materials are already being utilized in the fabrication of commercial heat recovery systems.

Three heat recovery schemes have been identified as possessing outstanding merits: the direct-contact condensing heat exchanger, the indirect-contact condensing heat exchanger, and the conventional economizer. An assessment of each of these schemes has led to the following conclusions:

1. A heat recovery system installed on a boiler rated at 20 MBtu/hr can have an SIR ranging from 2 to 6 depending on the type of heat recovery system. The SIR for larger boilers will be larger, and smaller for smaller boilers.

2. Gas-fired boilers are prime candidates for flue gas heat recovery. The combustion products of natural gas are clean and contain a large amount of water vapor and, hence, latent heat. A condensing heat exchanger is, therefore, the natural choice to recover this energy.

3. An indirect-contact condensing heat exchanger is, in general, the most favorable heat recovery scheme. It can be used to preheat make-up water and combustion air for both gas- and oil-fired boilers. If the temperature of the fluid being preheated is reasonably low, efficiency improvements can be high due to the capture of latent heat. An efficiency improvement of 5 to 8 percent can be attained with a SIR of between 3 and 4. Figure 36 illustrates the conditions under which these heat exchangers should be considered as candidates.

4. A direct-contact heat exchanger is limited by its low temperature level; it is suitable for natural-gas- and light-fuel-oil-fired boilers or for general process heating. Direct-contact heat exchangers should not be used for boiler efficiency improvement unless boosted by a heat pump or unless heat recovered from one boiler can be used to preheat multiple boilers.

5. The conventional economizer can be used with natural gas, oil, and coal, but its heat recovery capability is limited by controls designed to avoid corrosion due to condensation. With economizers, boiler efficiency improvement is independent of the percentage of makeup water since all feedwater is preheated anyway. Economizers should be considered for stack gas temperatures above 450 °F, if the boiler is operating more than 2,500 hours per year. A conventional economizer has a large SIR.

6. Because of many factors affecting heat recovery systems, installed cost of a heat recovery system is site-specific. It ranges between 2 to 5 times the equipment cost.

7. If boiler stack gas energy cannot be fully used to improve boiler efficiency, preheating process air or water may be attractive using a direct-contact heat exchanger. The direct-contact heat exchanger can be

45

economically justified if a low temperature process load constitutes more than 10 to 15 percent of the fuel energy input to the boiler and the boiler operates more than 4,000 hours annually. Under these conditions, an SIR of over 6 is possible. Management and tempering of the recovered energy may be possible by judicious use of heat pumps.

8. Additional benefits of installing a heat recovery system may include reduction in pollution emission, increased compactness of flue-gas particulates for easier removal, and reduced stack height and diameter.

RECOMMENDATIONS

To prove the economics and reliability in Navy Environments of the various proposed schemes for recovering waste heat from stack gas, the following RDT&E recommendations are made:

1. A flexible and versatile test-boiler should be designed and built, and various heat-recovery schemes should be subjected to test and evaluation under a variety of conditions.

2. The test matrix should simulate energy output from the heat recovery in the form of hot air or hot water, suitable for improving boiler efficiency and for a variety of process heating applications at various boiler loads.

3. A series of CAE software packages should be developed to calcu-late detailed estimates of recoverable heat, economic benefits, size and operating conditions of the heat exchanger, and desirability of heat pumps, with minimum amount of input information and a minimum amount of decisions and choices to be made by the designer. The software will cover both practical categories of heat recovery: to preheat feedwater or combustion air; and to heat process air, water, or steam.

4. To continue from Recommendation 3, an expert system should be built for microcomputers that will require no decision and selection at all from the user. The entire engineering (including optimization, trade-off, and consultation of an experience-based data bank) would be done automatically, with a final recommendation unequivocally offered to the user upon receiving input parameters and limiting conditions.

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Table 1.   Portrait of Navy Stationary Boilers[a]

Item	Capacity Range, MBtu/H					
	Small Boilers[b]			Large Boilers[b]		
	<5	5-10	10^+-25	25^+-50	50^+-100	>100
All Boilers	63.6	15.3	8.7	5.0	4.3	3.1
Total Capacity	10.3	7.3	10.1	11.9	28.3	32.1
Fuel Consumption:						
Oils	5.5	6.7	9.0	9.8	23.3	45.7
Gas	9.5	6.1	8.8	17.6	36.1	21.9
Coal	0.1	0	0	0	28.6	71.3
Avg. Capacity Factor[c]	16.5	22.2	22.1	25.8	25.0	31.3

[a] Based on data on 2,012 boilers out of the 2,902 extracted from NAPSIS. Fuel consumptions are based on FY84 figures.

[b] All figures are in percentage points.

[c] Average capacity factor $= \dfrac{\text{Annual fuel consumption (MBtu/yr)}}{\text{Boiler rated capacity (MBtu/H)} \cdot 8,760 \text{ (H/yr)}}$

This is a measure of the extent a boiler is utilized.

Table 2. Ultimate Analysis of Typical Fuels

Species	Percentage Weight by Species for Fuel Types			
	Coal	Coal-Water Slurry	No. 2 Fuel Oil	Natural Gas
Carbon	75.10	55.30	87.00	74.70
Hydrogen	4.80	3.60	12.50	23.30
Nitrogen	1.26	1.01	0.00	0.76
Oxygen	5.96	4.39	0.00	1.22
Sulfur	0.76	0.60	0.50	0.00
Moisture	6.20	30.70	0.00	0.00
Ash	5.90	4.40	0.00	0.00
High Heating Value, (Btu/lb)	13,380	9,910	19,520	22,904

Table 3. Evaluation of Heat Recovery Schemes to Improve Boiler Efficiency*

Item	Weighting Factor	Run-around Coil	Conventional Economizer	Direct-Contact	Indirect-Contact Condensing	Plate Heat Exchanger (Air/Air)	Heat Wheel	Heat Pipe
Cost Effectiveness	0.25	3	8	2	7	5	2	4
Energy Savings	0.20	3	4	3	7	3	3	3
Reliability	0.10	6	7	6	6	6	3	3
O&M	0.15	6	7	4	6	8	3	7
Ease of Retrofit	0.15	5	8	7	7	4	3	4
Compact Size	0.10	8	9	5	5	7	4	7
Pollution Control	0.05	1	1	8	5	1	1	1
Average Ratings:		4.45	6.70	4.25	6.45	5.00	2.75	4.30

Rating scale: 1 = unsatisfactory; 10 = outstanding.

*Heat pumps (being supplementary) and organic Rankine cycle (being uncertain in its practical merit) are excluded.

51

Table 4. Typical Design Parameters for a Conventional Economizer

Fuel Type	Minimum Inlet Water Temperature (°F)	Minimum Exit Flue Gas Temperature (°F)	Fin Density, n (fins/in)	Fin Thickness, b (in)	Tube Diameter, D_o (in)	Fin Height, h (in)
Natural Gas	210	300	5	0.060	2.0	0.875
No. 2 Fuel Oil	220	325	4	0.060	2.0	0.750
No. 5 and No. 6 Fuel Oils	240	350	3	0.075	2.0	0.750
Coal	240	350	2	0.105	2.0	0.750

Table 5. Summary of Actual Applications of Fiberglass Reinforced Plastic
(FRP) in Corrosive Flue Gas Desulfurization Service

Equipment	Corrosive Environment
Venturi Cooler	160 °F boiler flue gas.
Stacks and Chimney Liners	Flue gas at 350 to 375 °F with peaks to 500 to 700 °F.
Circulating Tanks and Piping	Acidic fluids up to 180 °F.
Fans and Blowers	350 °F flue gas.
Scrubbers	300 to 350 °F bagasse flue gas, trash burner flue gas.
Electrostatic Precipitator Vibrator Housing	H_2S and SO_2 fumes at 280 to 360 °F.
Mist Eliminators and Grating Support Beams	Sulfurous acid to 350 °F and thiosorbic lime.
Desulfurizing Units	H_2S, SO_2, and citric acid.

Table 6. Comparative Evaluation of Heat Recovery Schemes Based on
Preheating Process Fluid

Item	Weighting Factor	Heat Recovery Scheme					
		Run-around Coil	Direct-Contact	Indirect-Contact Condensing	Plate Heat Exchanger (Air/Air)	Heat Wheel	Heat Pipe
Quickness of Payback	0.25	5	9	8	6	4	6
Energy Savings	0.20	6	9	8	7	7	7
Reliability	0.10	6	6	6	6	3	3
O&M	0.15	6	4	6	8	3	7
Ease of Retrofit	0.15	6	7	7	5	4	5
Compact Size	0.10	6	7	6	7	4	7
Pollution	0.05	1	8	5	1	1	1
Average Ratings:		5.50	7.40	7.00	6.20	4.20	5.75

Rating scale: 1 = unsatisfactory; 10 = outstanding.

54

Table 7. Heat Transfer/Pressure Drop Characteristics
of Direct-Contact Heat Exchangers

Parameter	Characteristics by Type of Exchanger		
	Spray Chamber	Baffle Plate	Packed Tower
Heat Transfer Performance / Unit Volume	Low	Medium	High
Pressure Drop / Unit Volume	Low	Medium	High
(inches of water)	(<0.1)	(0.5 to 1.0)	(5 to 10)
Hot Water Temperature (°F)	Low (100 to 110)	Medium (115 to 125)	High (130 to 140)
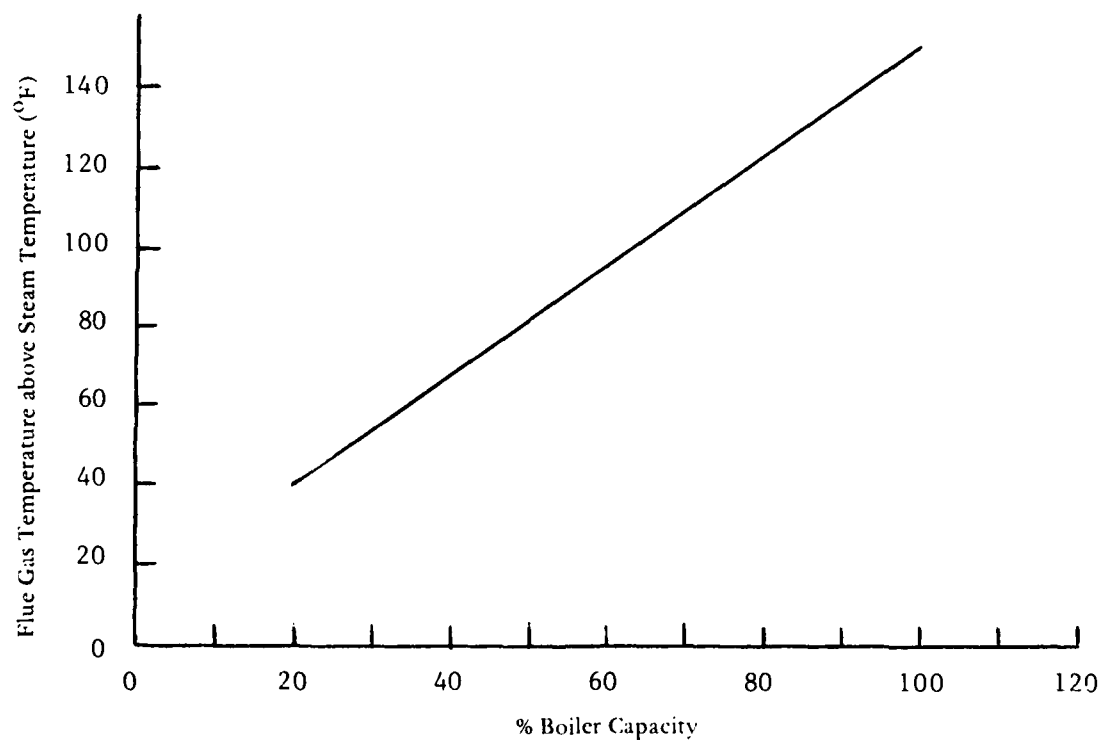
Figure 1. Typical flue gas temperatures above steam temperatures
at various boiler firing rates.

Typical
Boiler Losses
20%

Combustible Losses
0.5%

Radiation and
Convection Losses
1.5%

Blowdown Losses
1.0%

Dry Flue Gas
Losses

8.5%

Moisture in
Flue Gas
8.5%
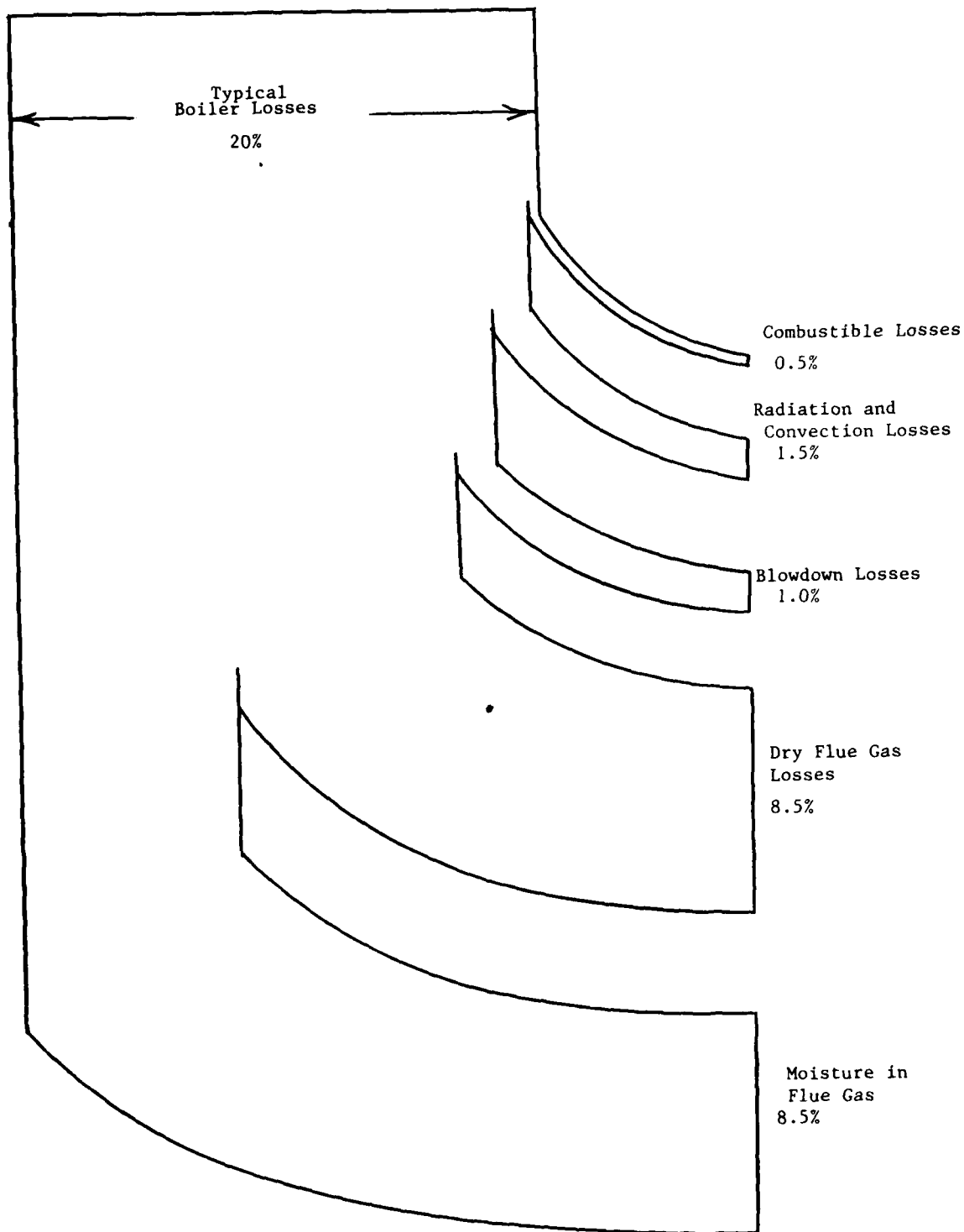
Figure 2. Distribution of typical boiler losses.

57

Oxygen

Nitrogen

Water Vapor

AIR SUPPLY

Carbon

Hydrogen

Sulfur

Oxygen

Nitrogen

Ash

Water

+   FUEL

Carbon Dioxide

Other Oxides:
CO, NO_x, SO_x

Oxygen

Nitrogen

Water Vapor

Refuse

Carbon

Ash

COMBUSTION PRODUCTS
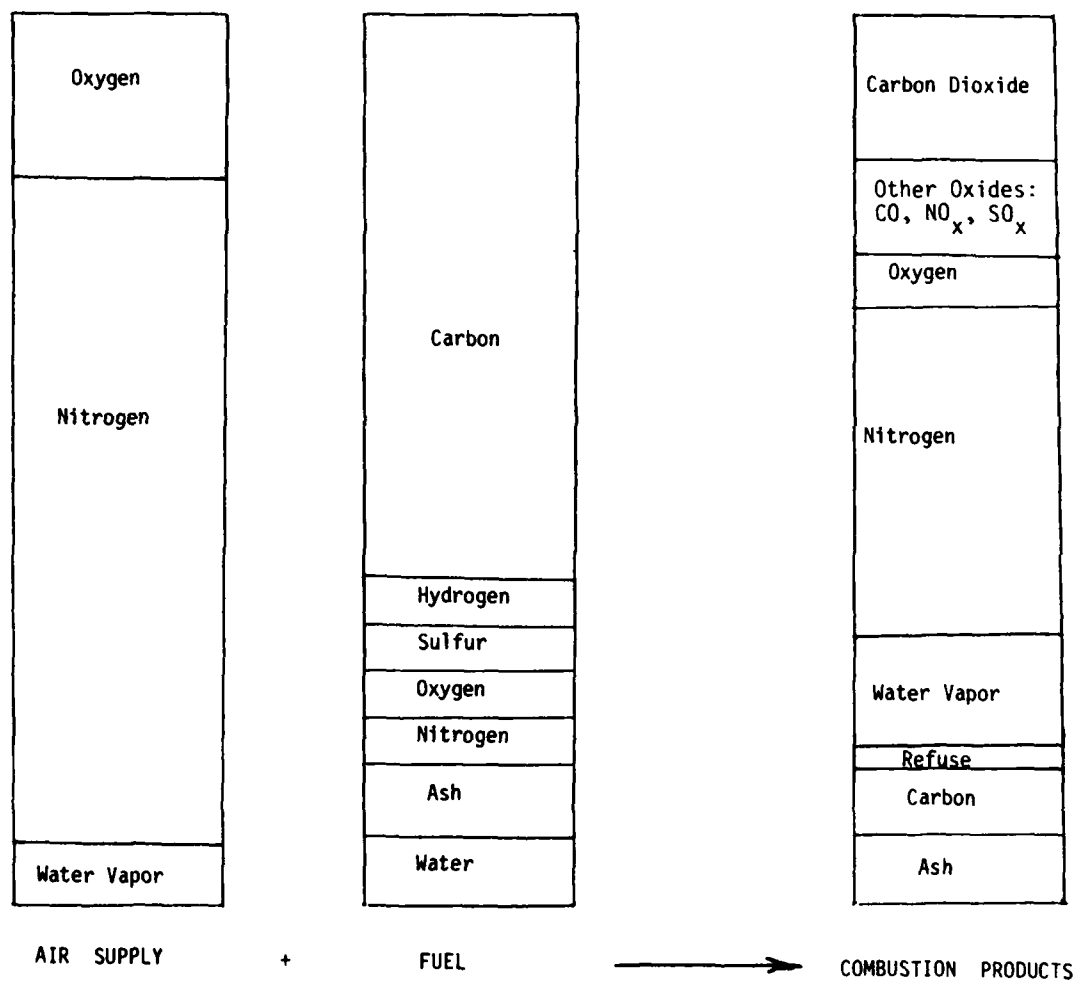
Figure 3.   The combustion process.
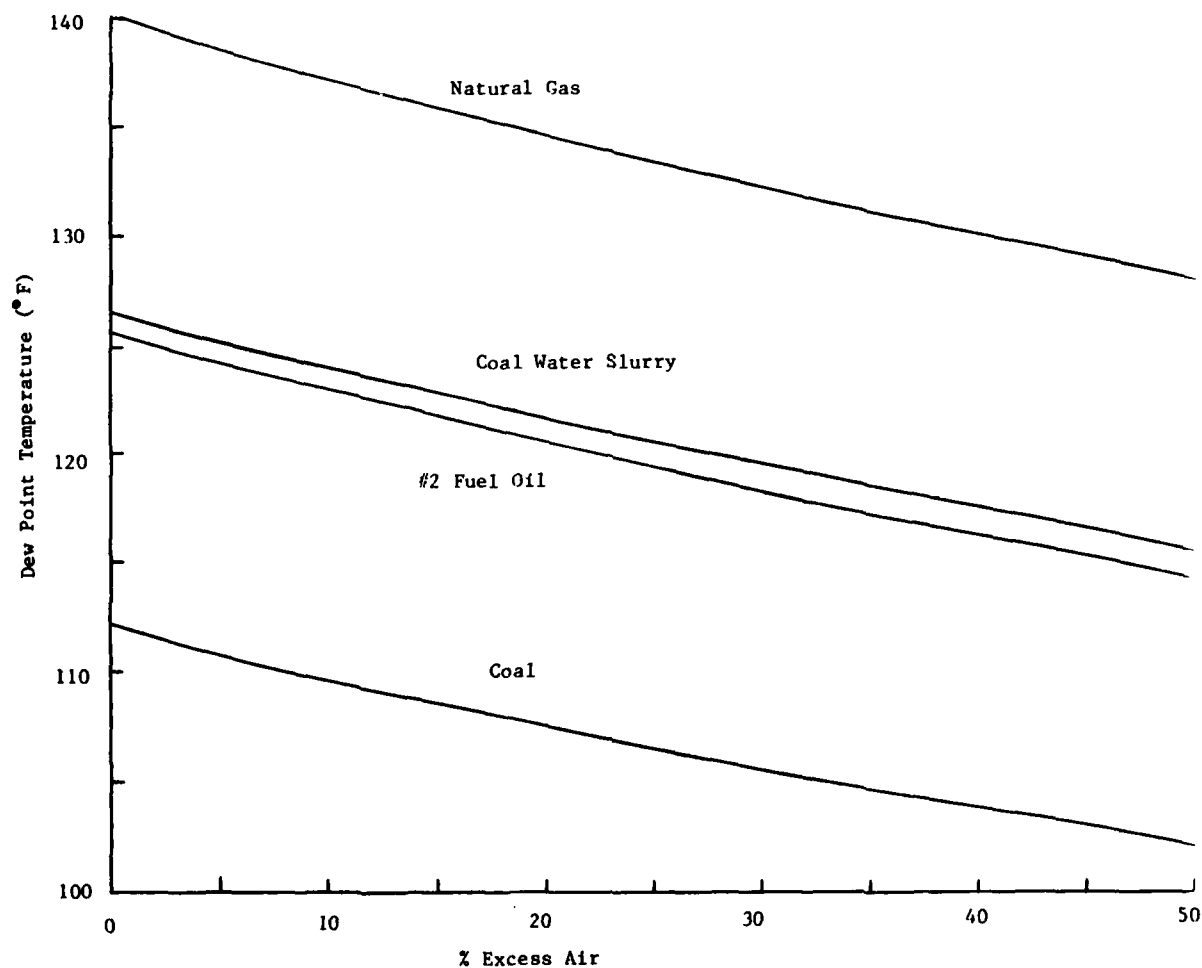
Figure 4. Relationship between water-vapor dew point of flue gas to excess air for various fuels.
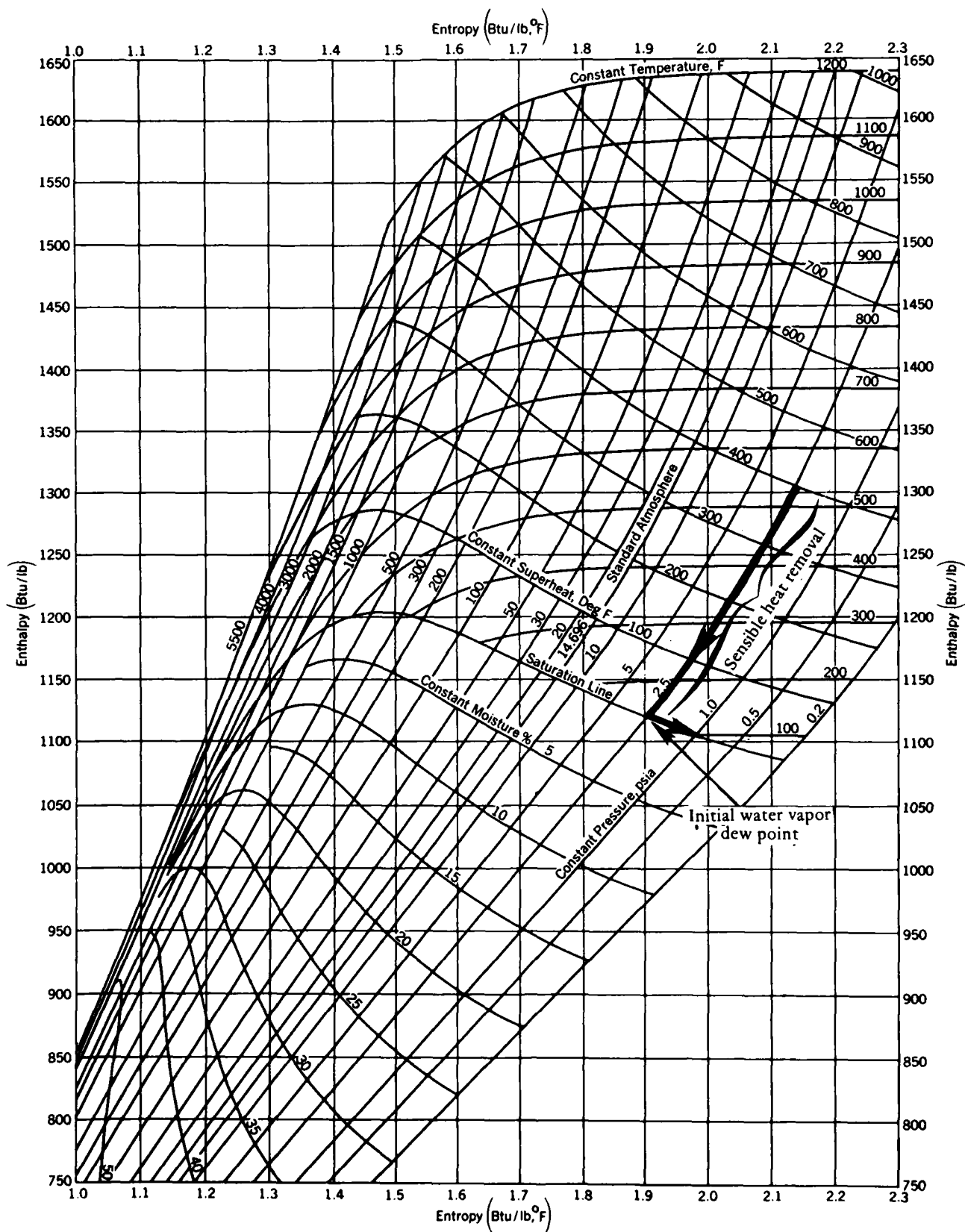
Figure 5. Enthalpy change of water vapor during cooling of flue gas.
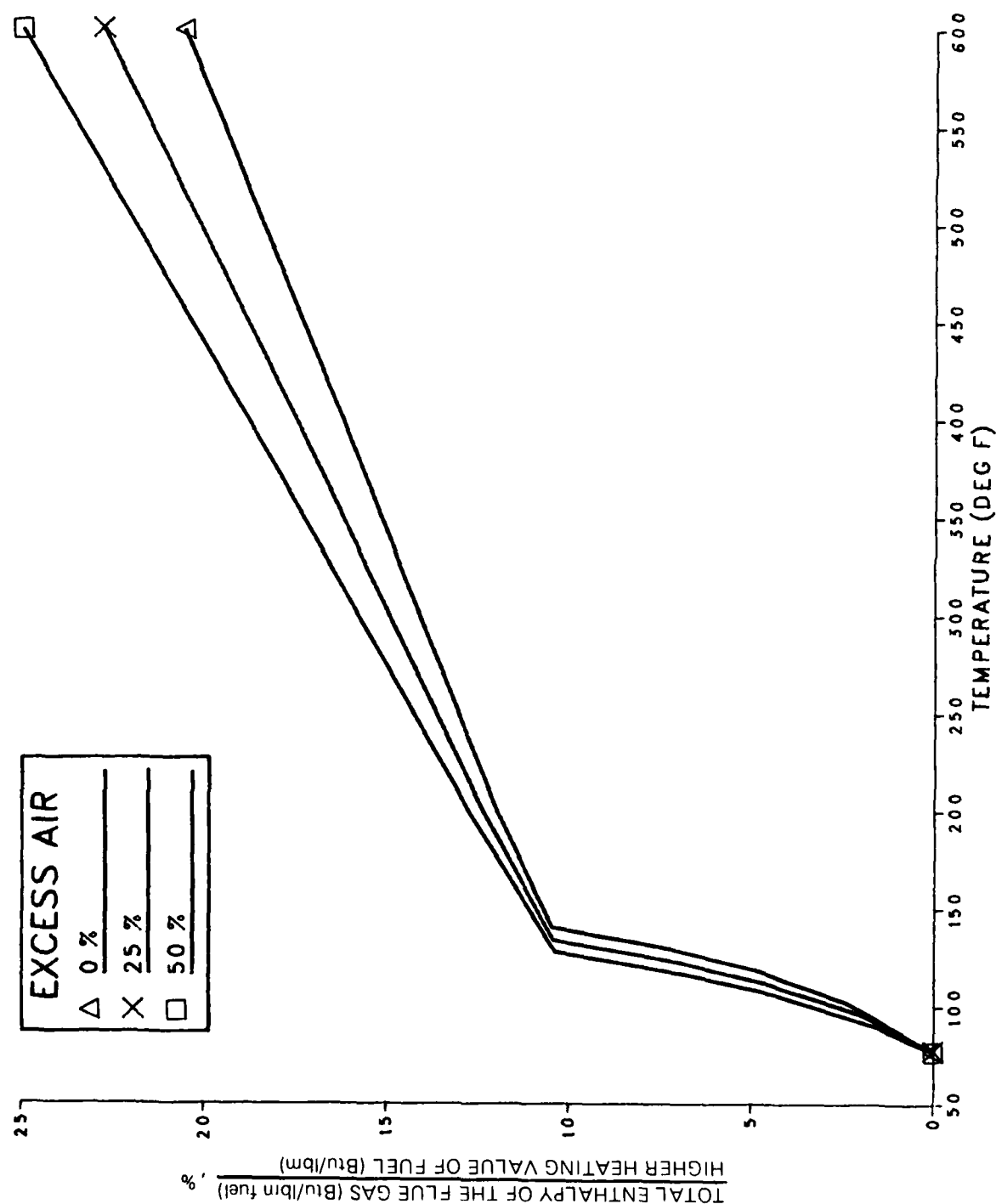
Figure 6. Enthalpy of combustion products as a function of temperature and excess air for natural gas.
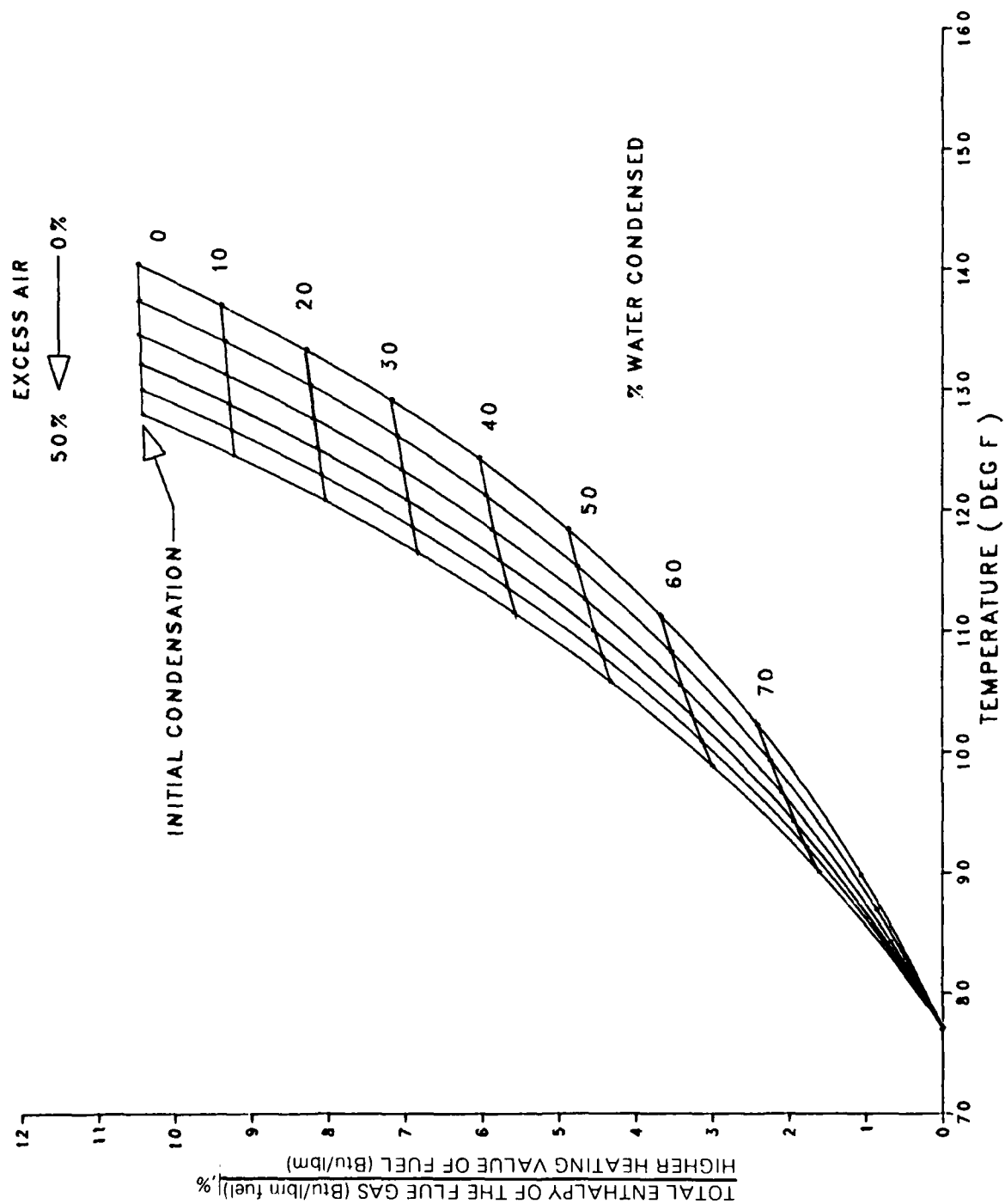
Figure 7. Enthalpy of combustion products during condensation as a function of temperature and excess air for natural gas.

62
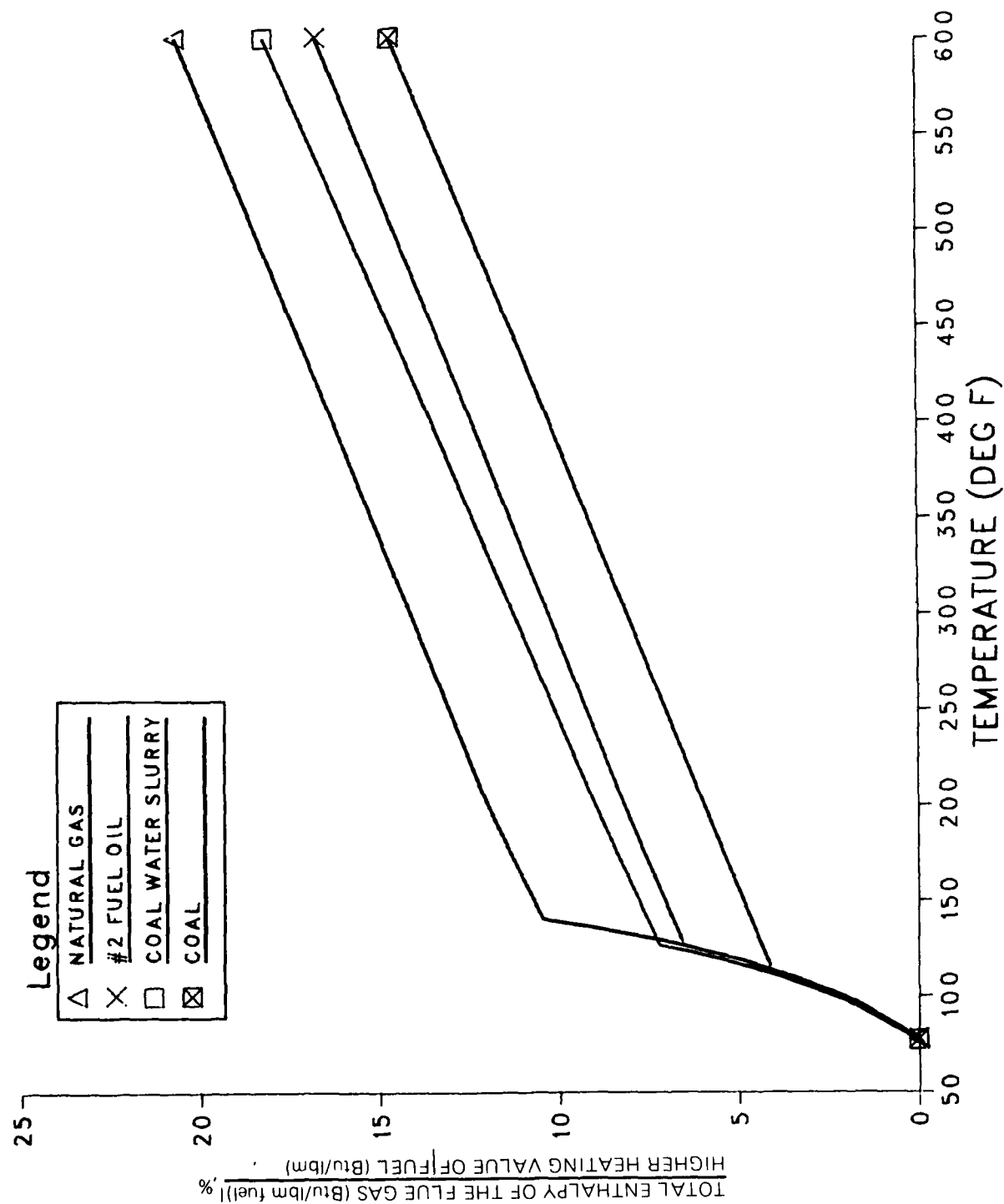
Figure 8. Flue gas enthalpy as a function of temperature for various fuels.
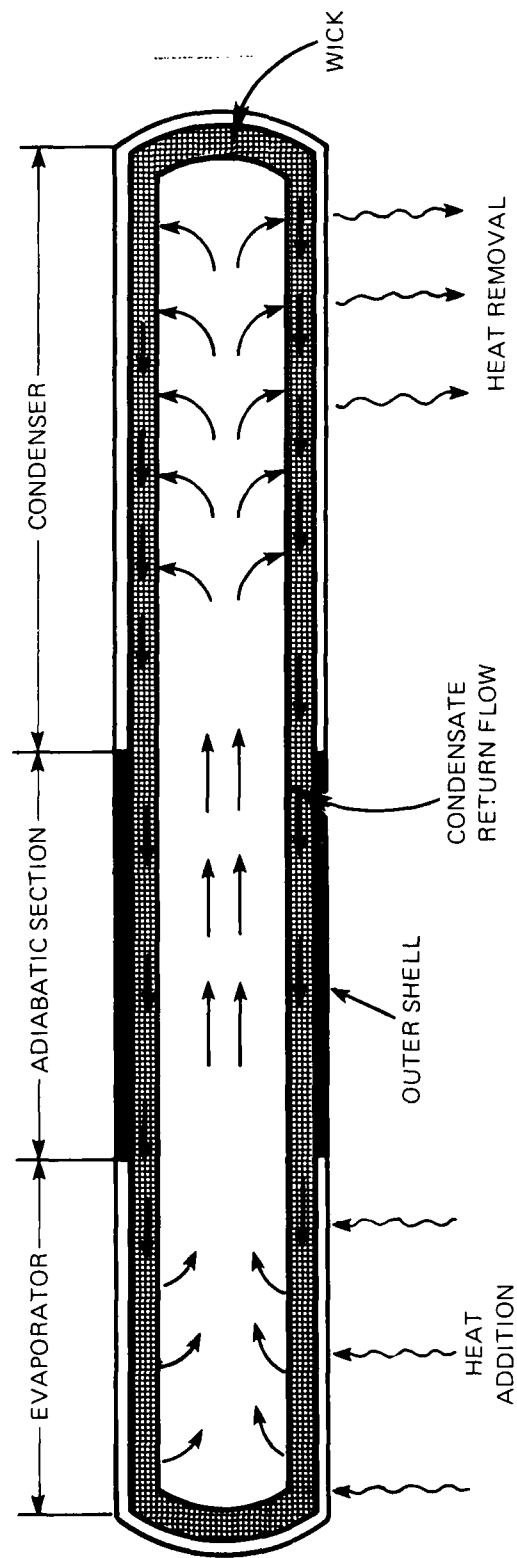
Figure 9. Components of a heat pipe.

64
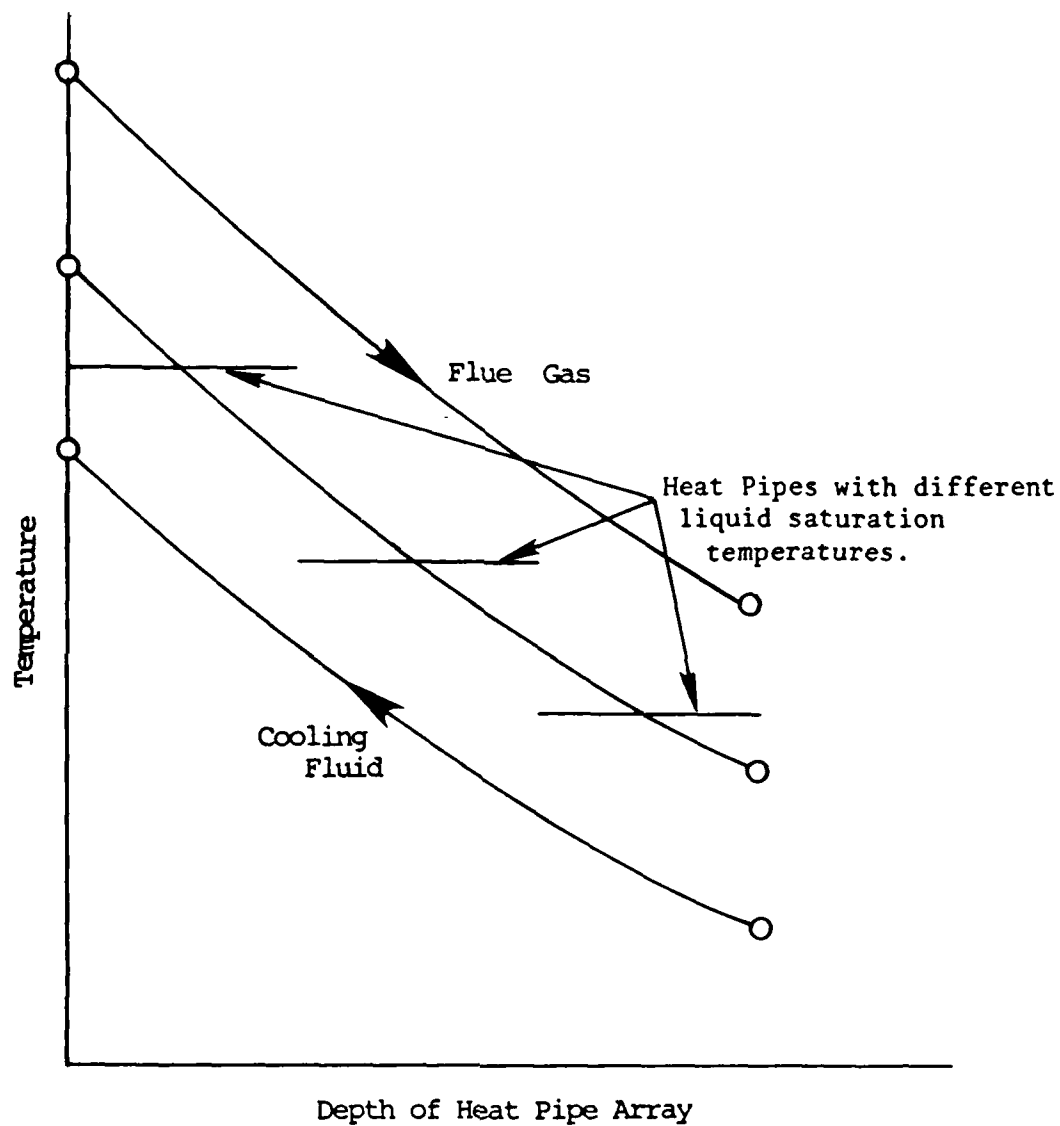
Figure 10. Heat pipe recovery scheme for removal of sensible heat.
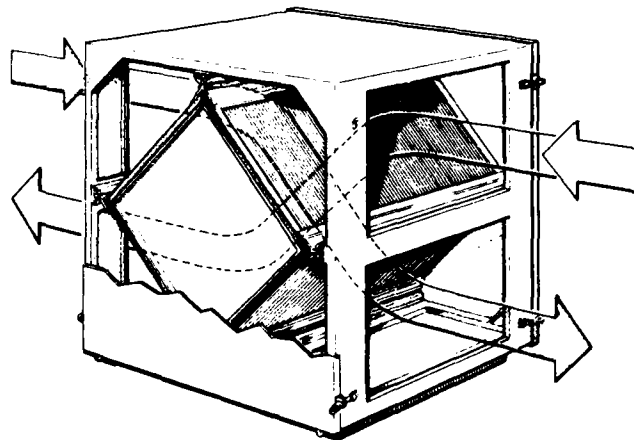
Plate heat exchanger for gases

Figure 11.   Plate heat exchanger for gases.
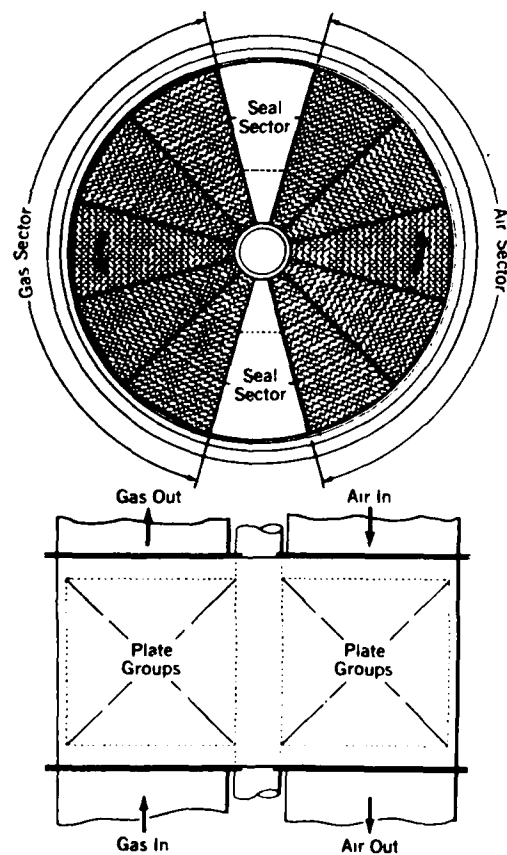
Figure 12.   Outline of rotary air heater.
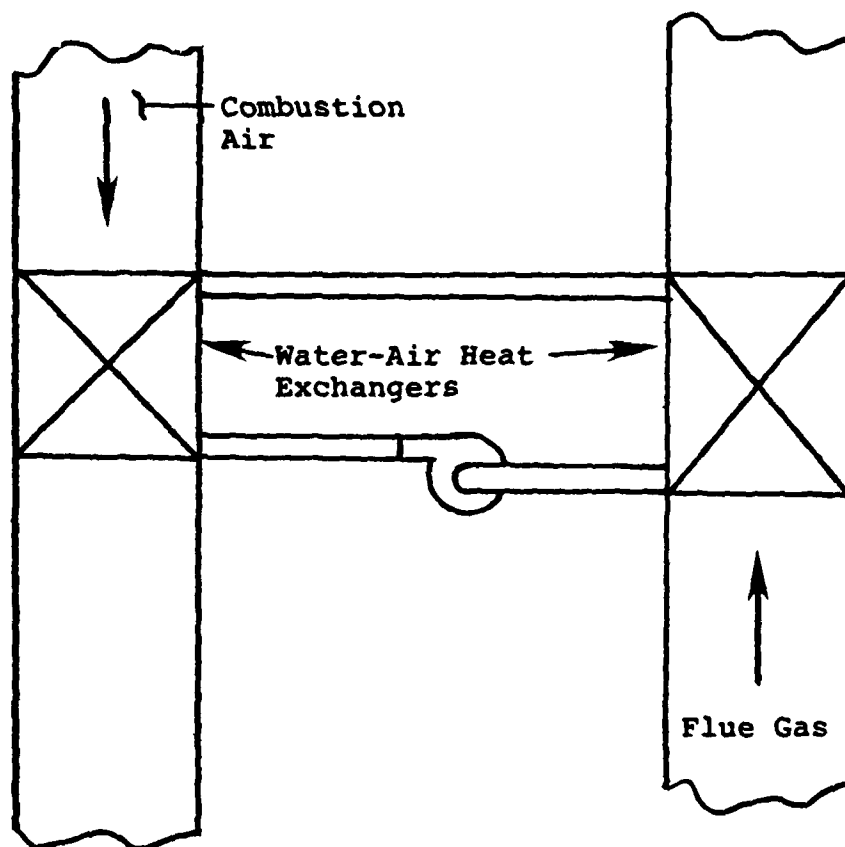
Figure 13. Run-around heat recovery system.
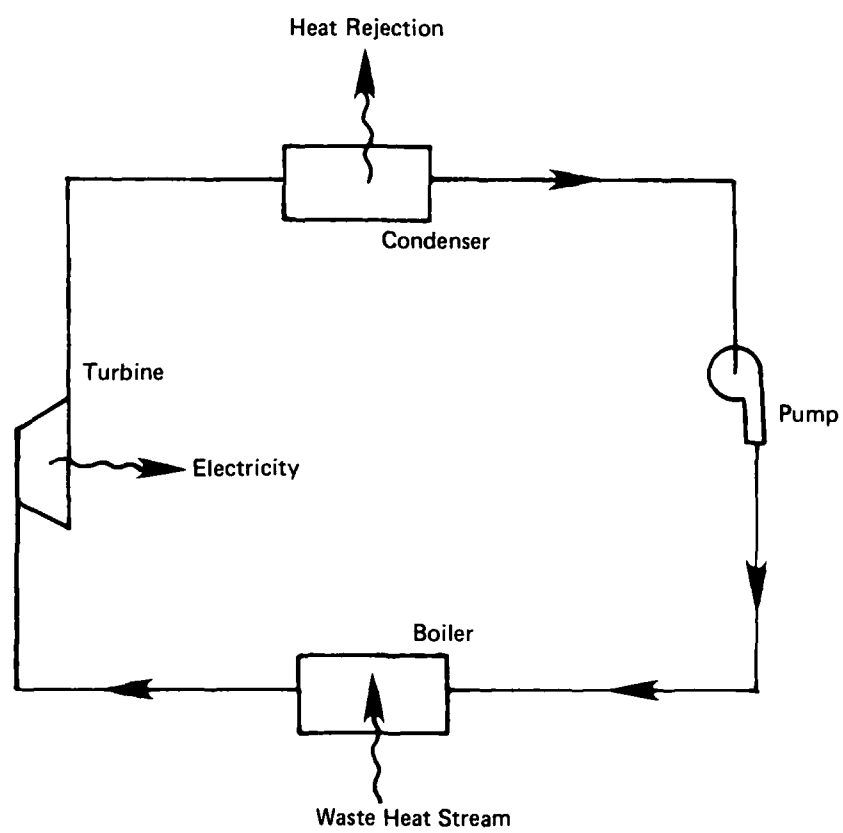
Figure 14. Organic Rankine cycle system.

Heat Use
(High Temperature Energy)

Condenser

Compressor

Electrical
Energy

Expansion Valve

Evaporator

Heat Source
(Recovered low temperature waste heat)
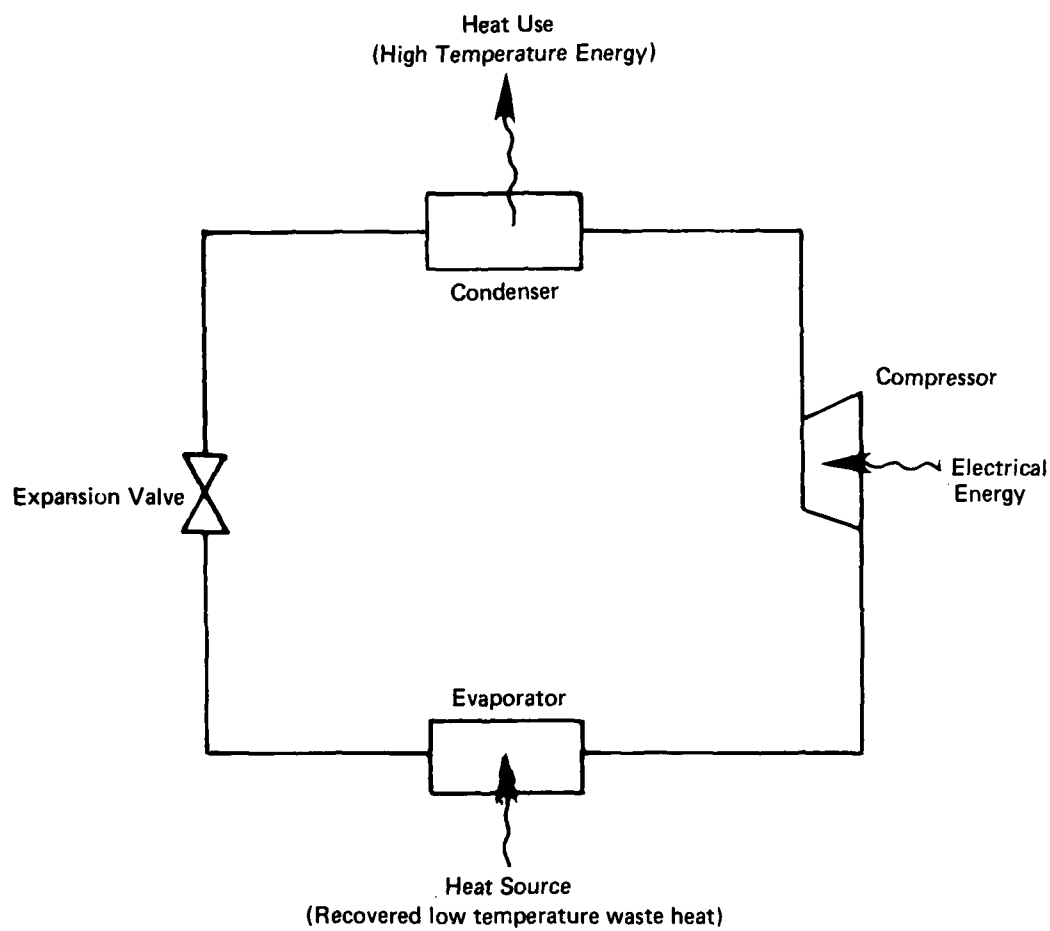
Figure 15. Closed-cycle vapor compression heat pump.
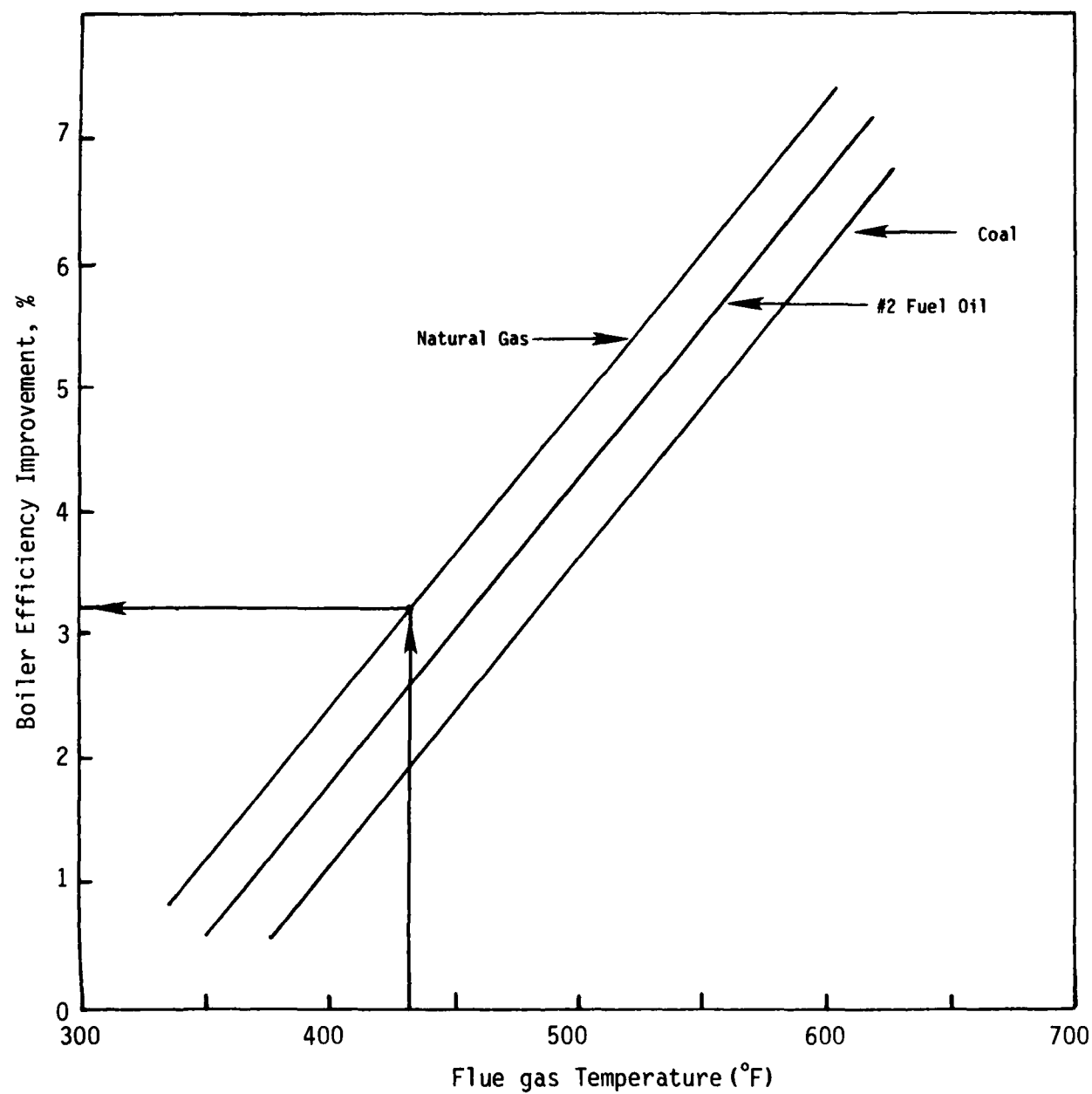
Figure 16. Approximate efficiency improvement of an
economizer as a function of flue gas
temperature and fuel type.
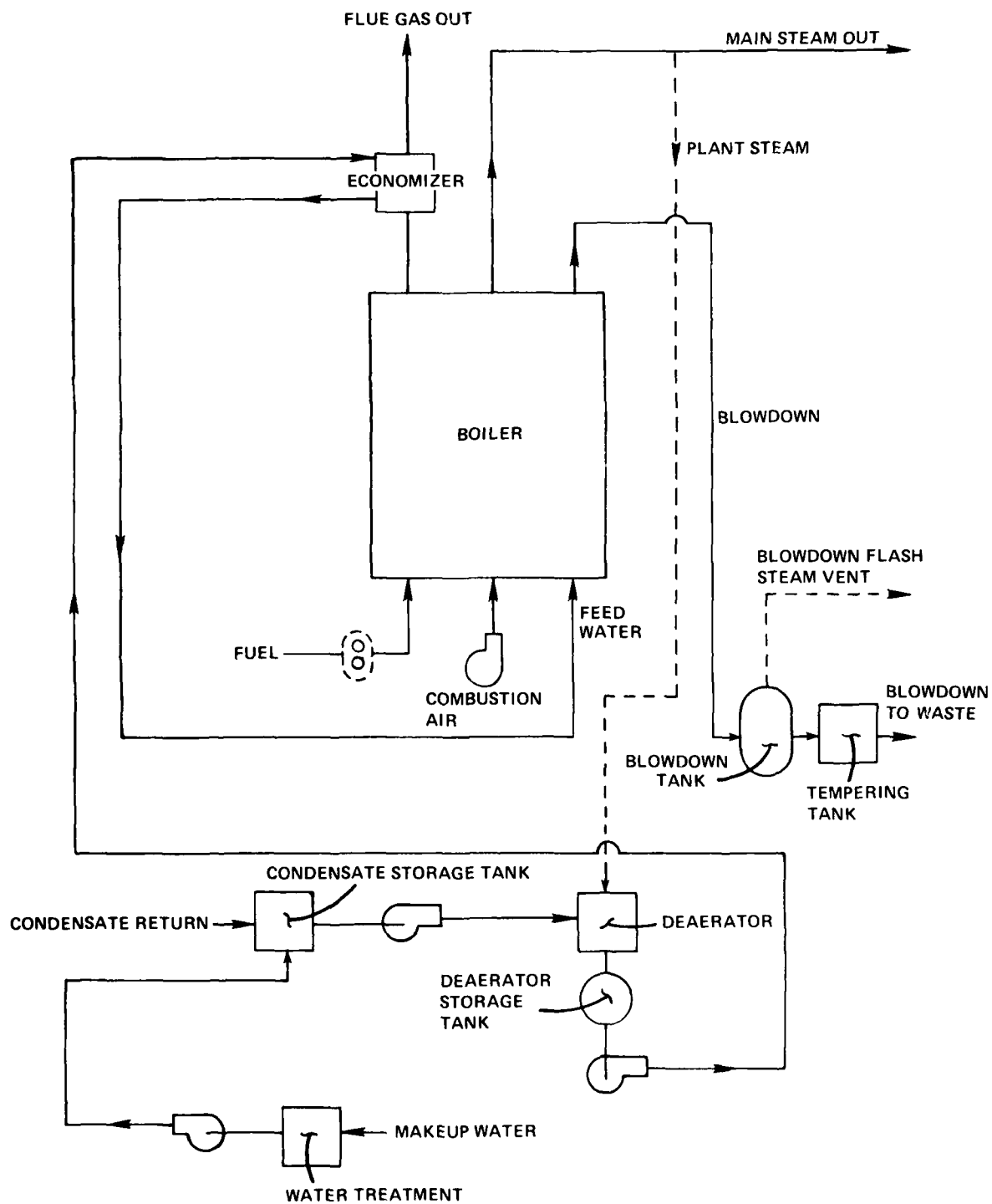
Figure 17. Boiler plant flow diagram with stack gas heat recovery economizer.
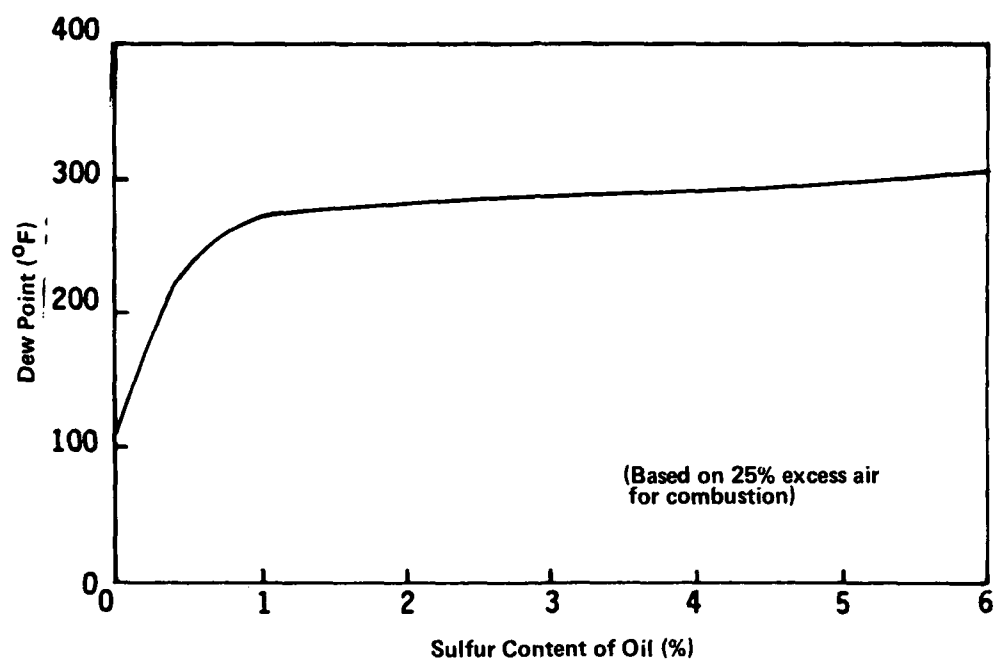
Figure 18.   Relationship of acid dew point to sulfur
content of oil.
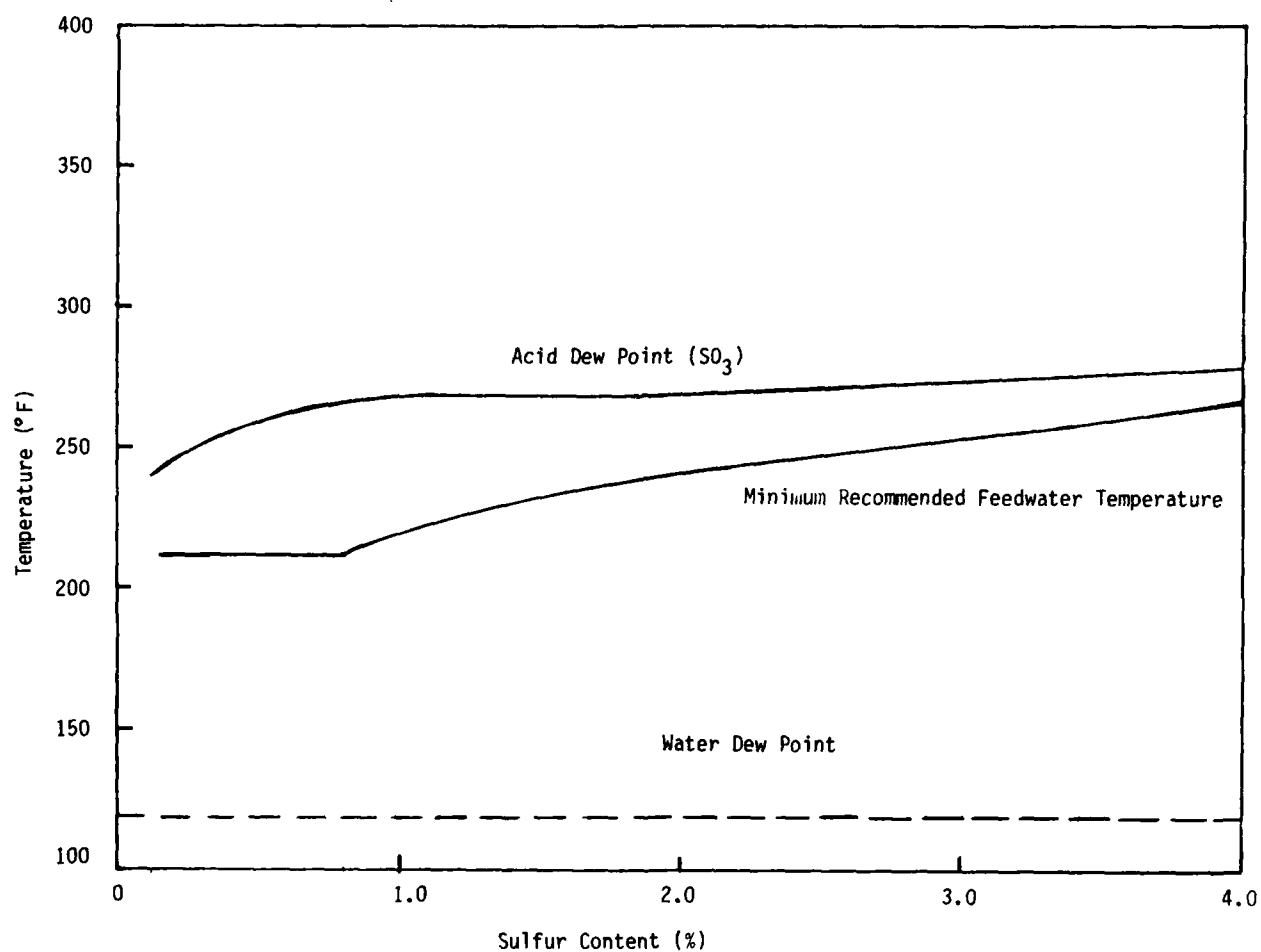
Figure 19. Minimum recommended feedwater temperature to avoid economizer tube corrosion.
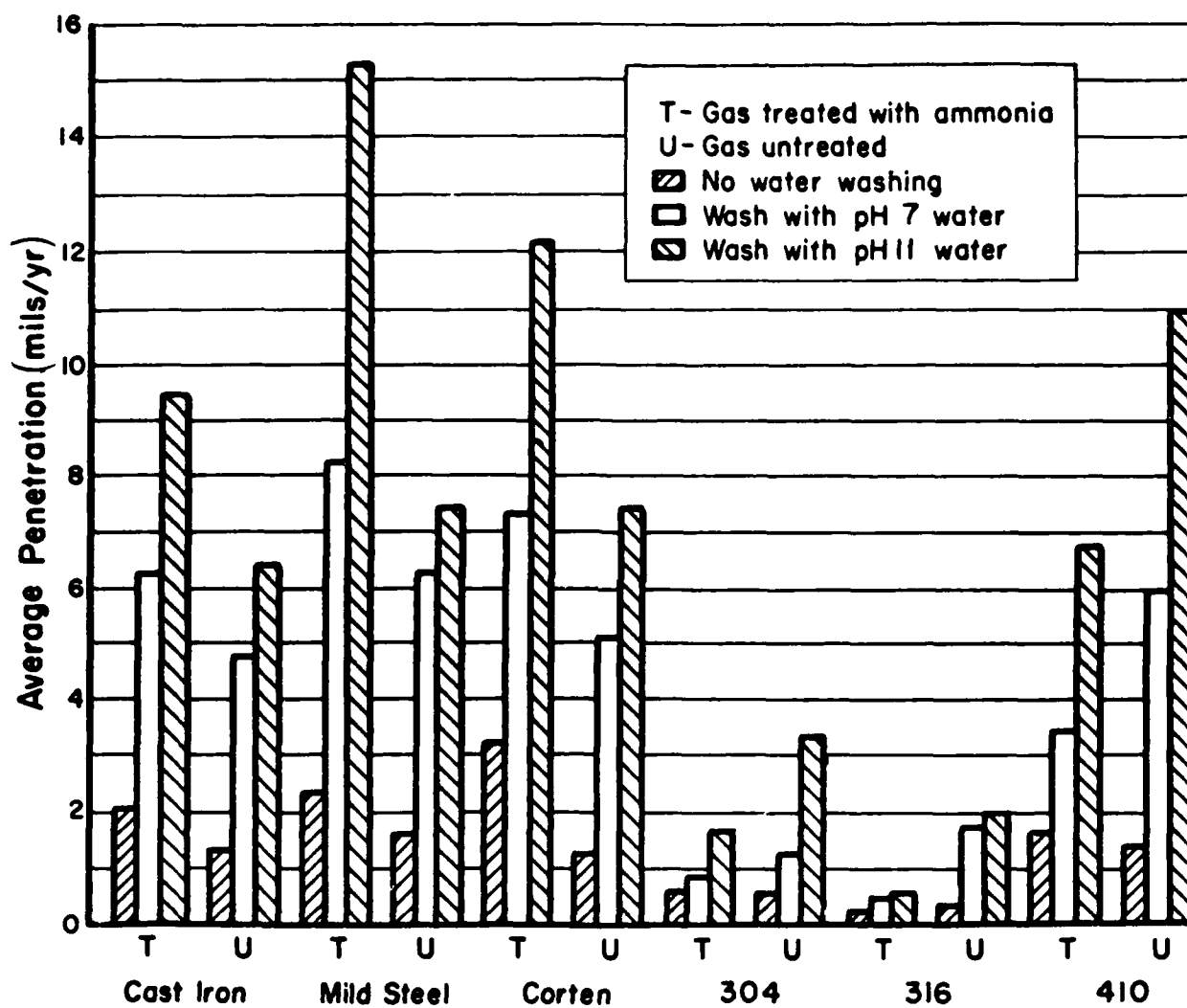
Figure 20. Rates of penetration in alloys exposed to low-temperature flue gas.
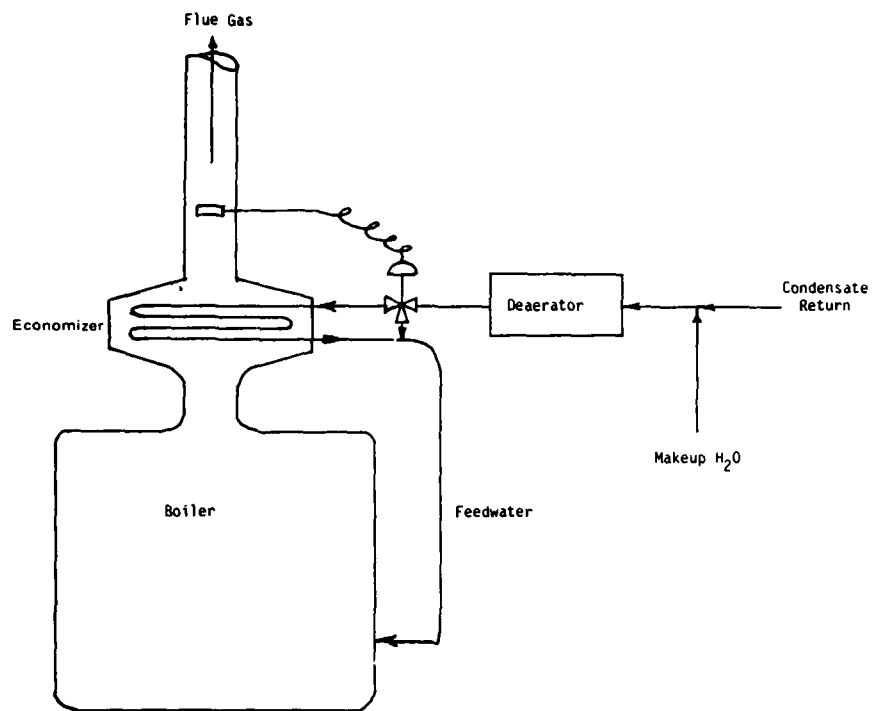
Figure 21. Economizer corrosion control by feedwater bypass.
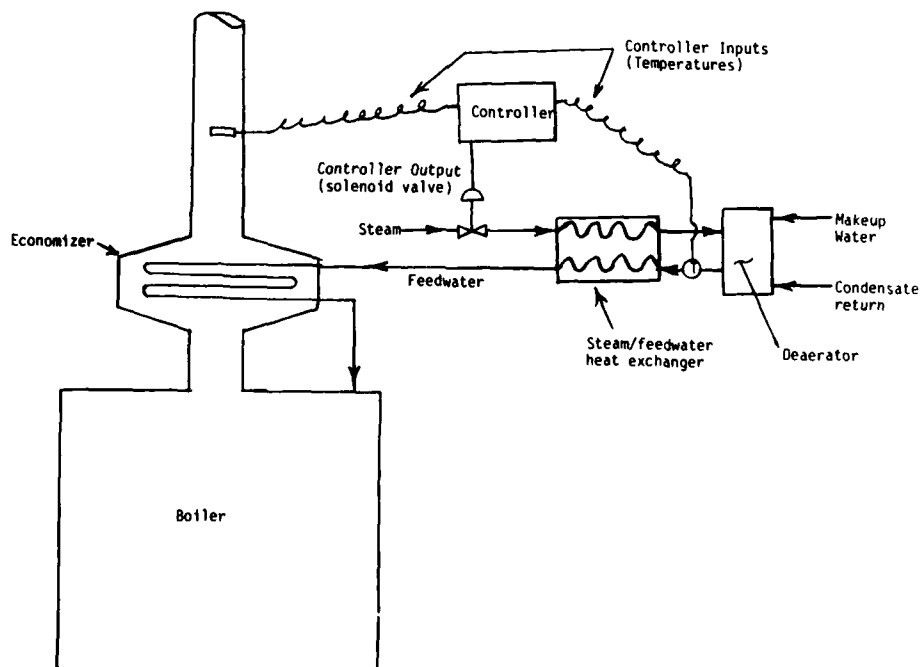
Figure 22. Economizer corrosion control by auxiliary steam pre-heat.
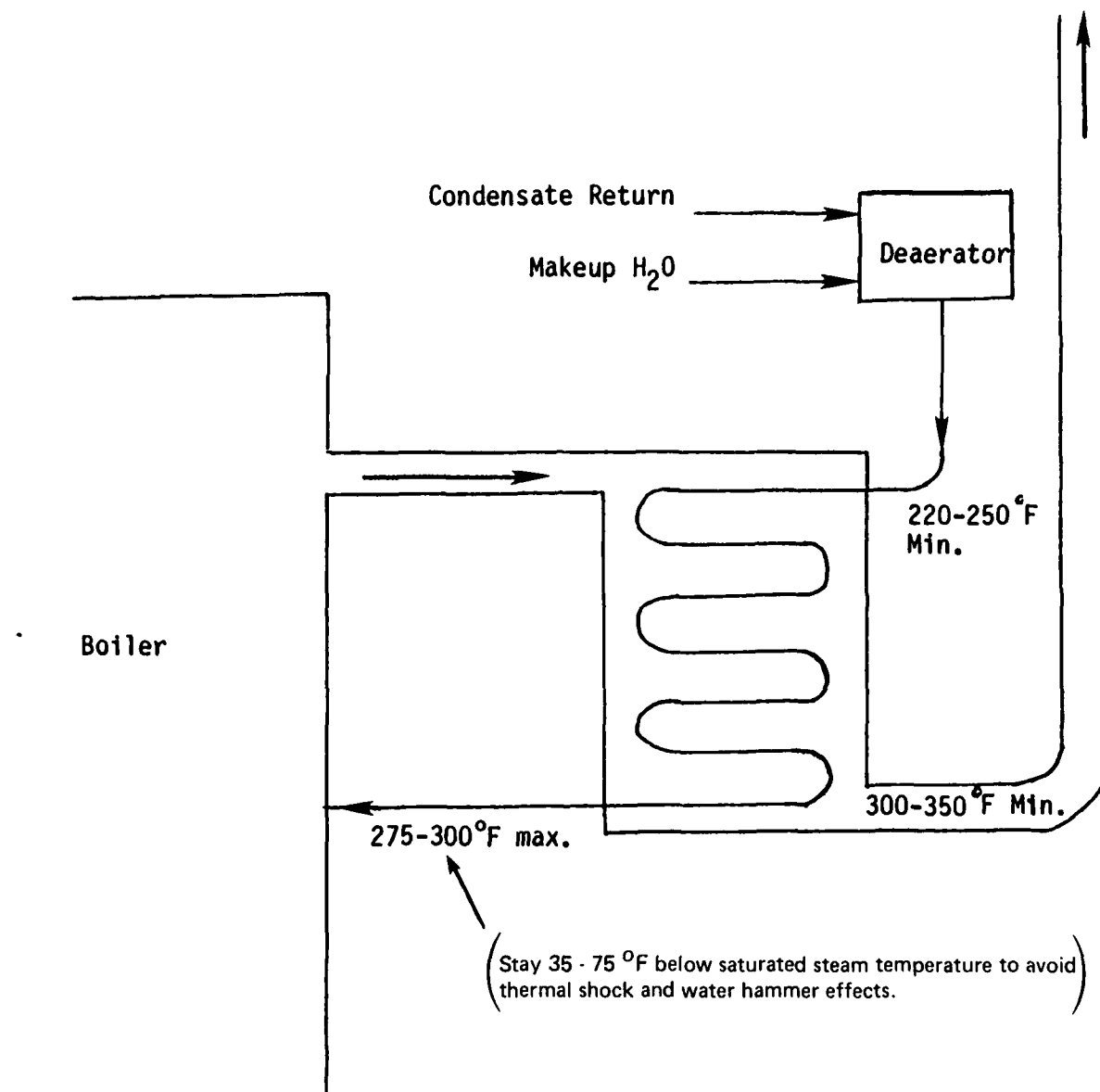
Figure 23. Design limitations of conventional economizer.
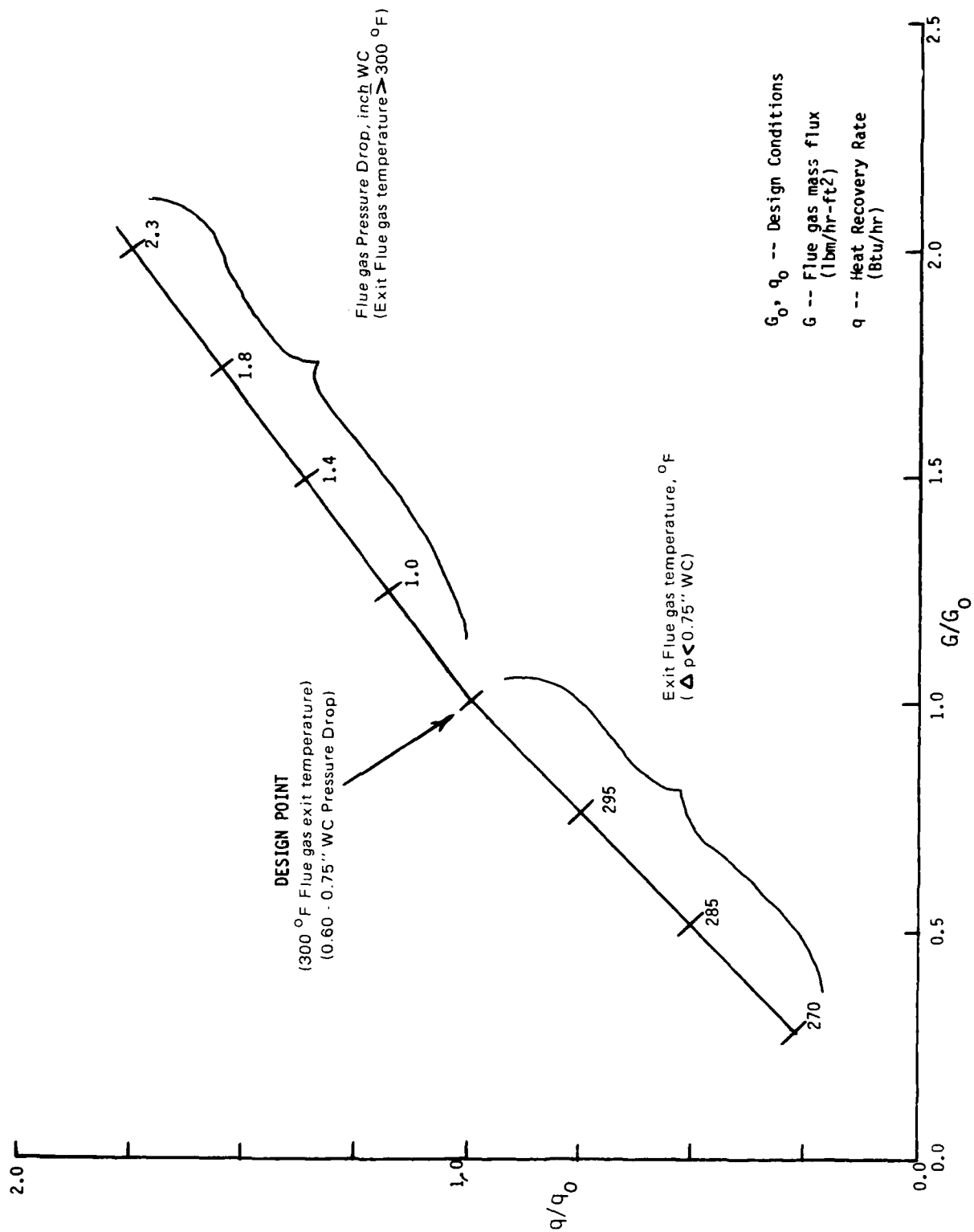
Figure 24. Effect of off-design conditions on a conventional economizer.

Conventional Economizer
(Contacting Surface: Finned Cast Iron Tubes)

Indirect Contact Condensing Heat Exchanger
(Contacting Surface: Teflon Coated Copper Tubes or Glass Tubes)

Direct Contact Heat Exchanger
(Contacting Surface: Water Droplets)
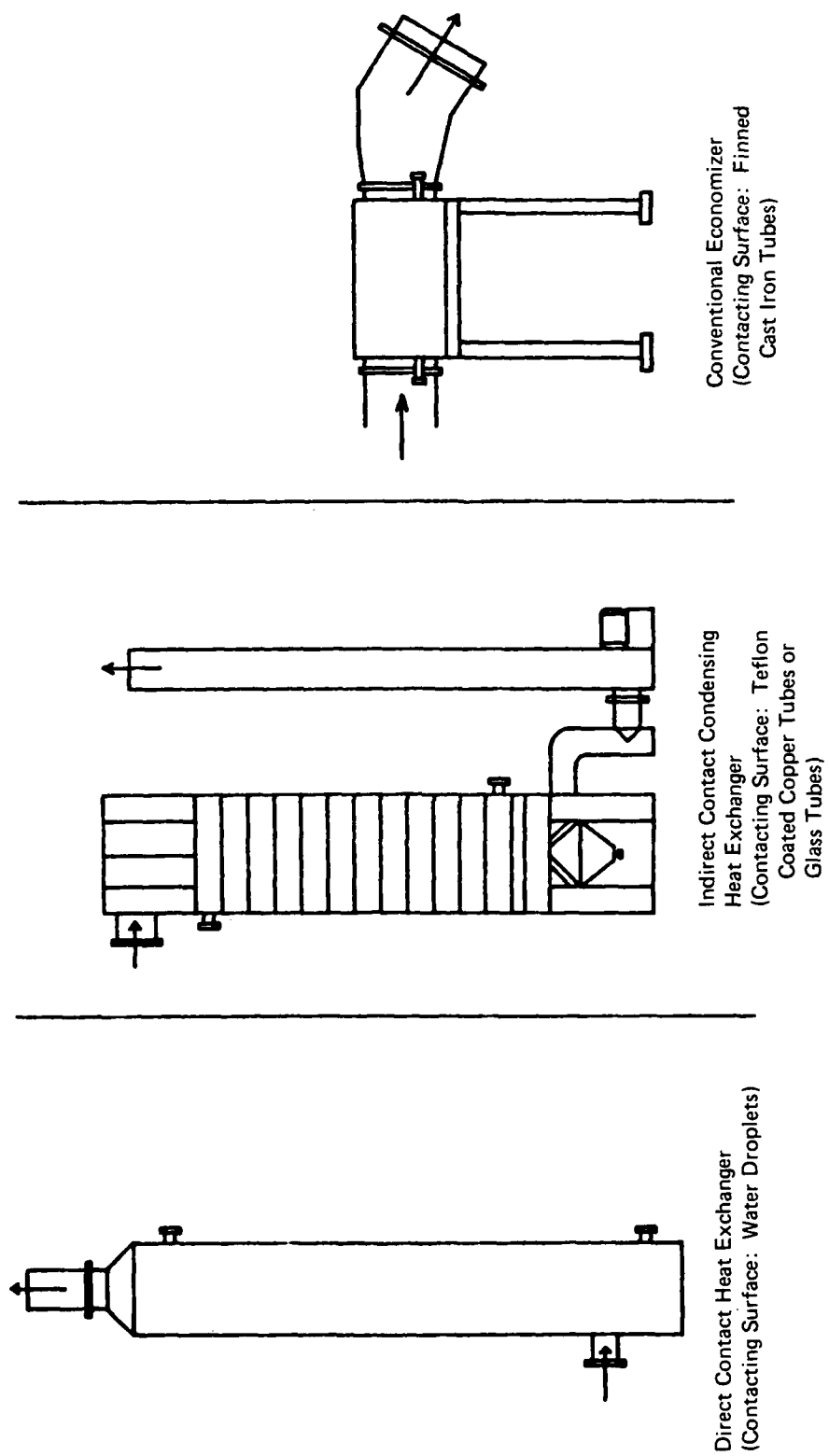
Figure 25. Relative size of the three heat recovery systems.

78
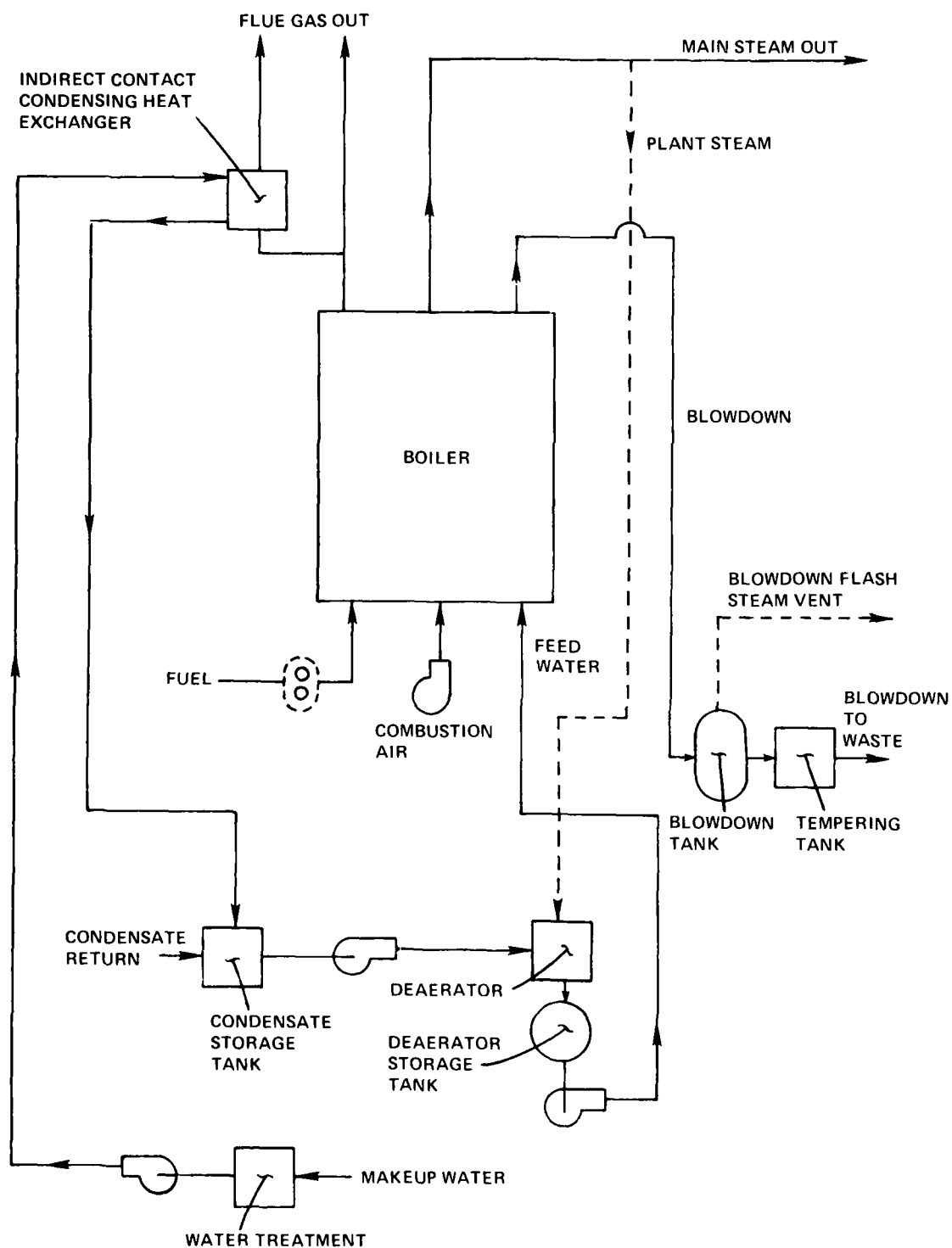
Figure 26. Stack gas heat recovery: makeup water is preheated by an indirect-contact condensing heat exchanger.
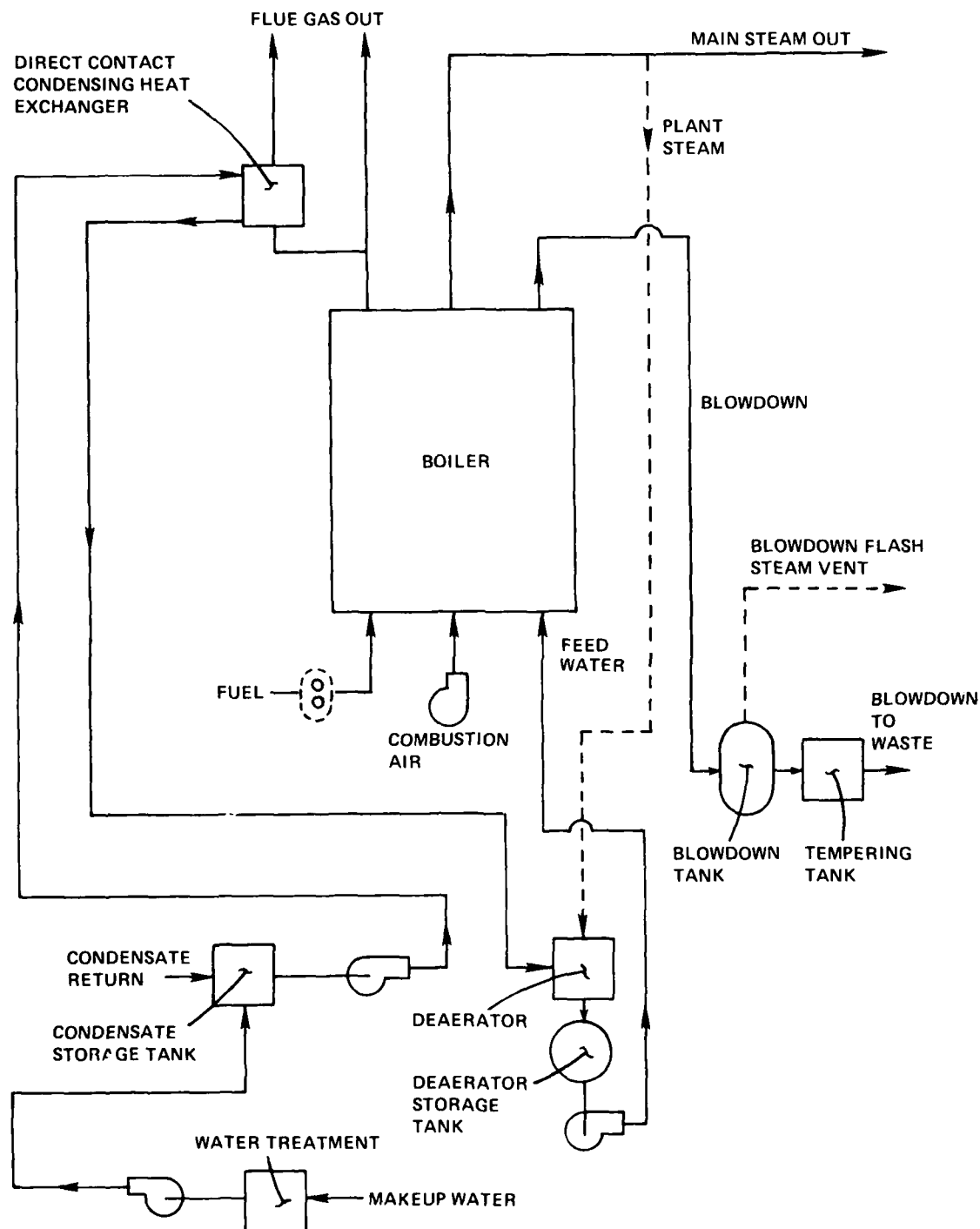
Figure 27. Stack gas heat recovery: feedwater is preheated by an indirect-contact condensing heat exchanger.
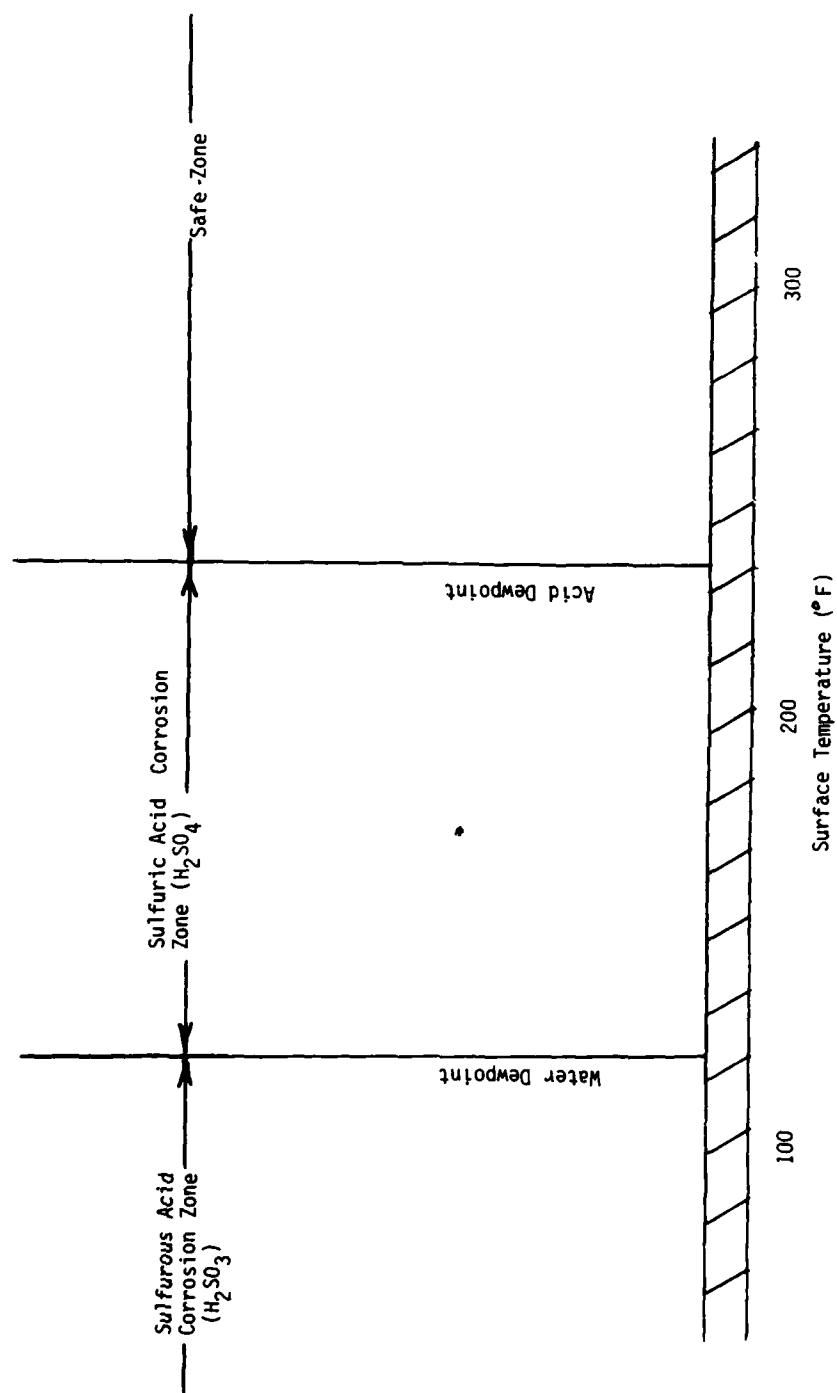
Figure 28. Typical flue gas corrosion zones.

Figure 29. The corrosion rate of various materials by sulfuric acid.
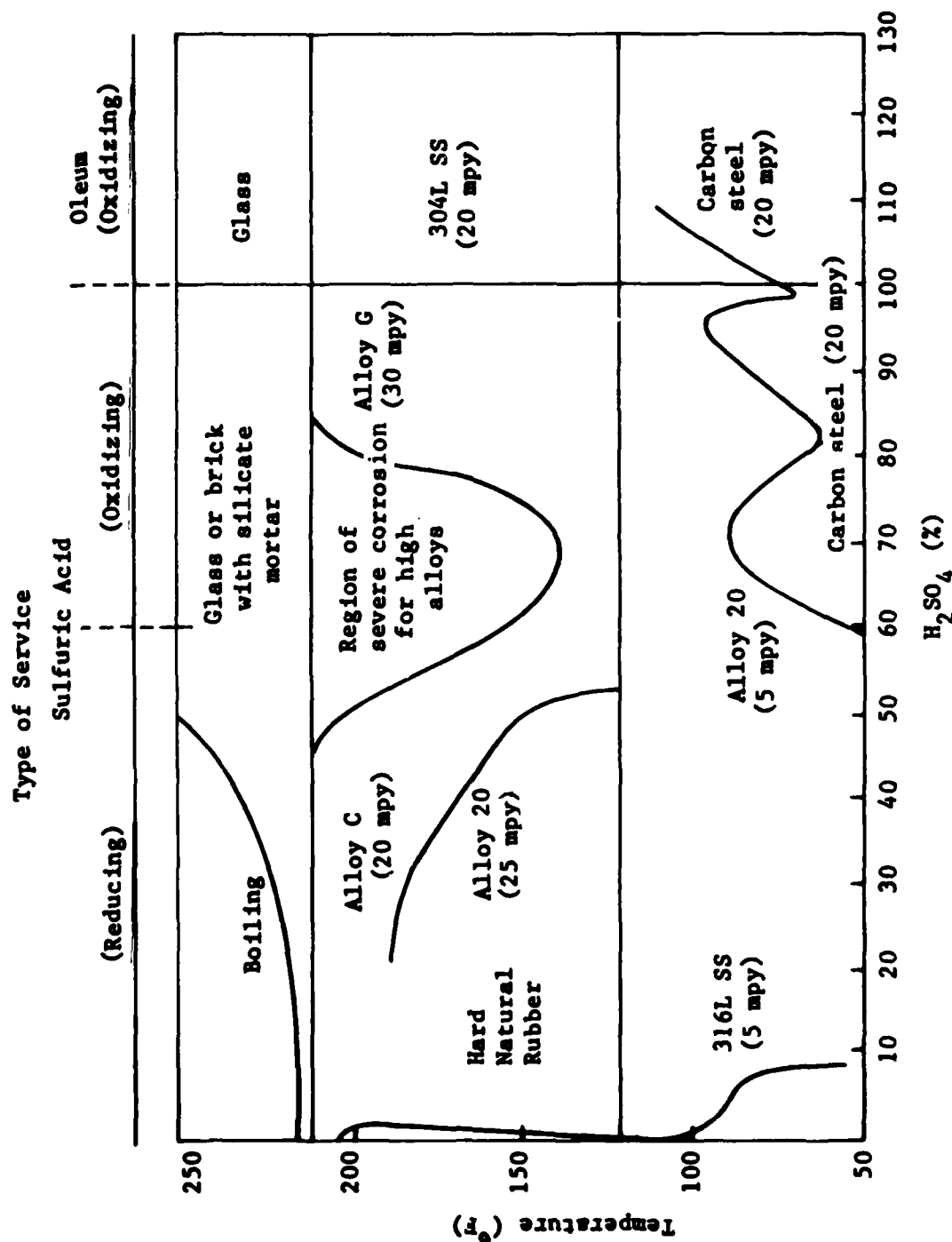
* Oleum is fuming sulfuric acid
* Corrosion rate--mpy (mils per year)
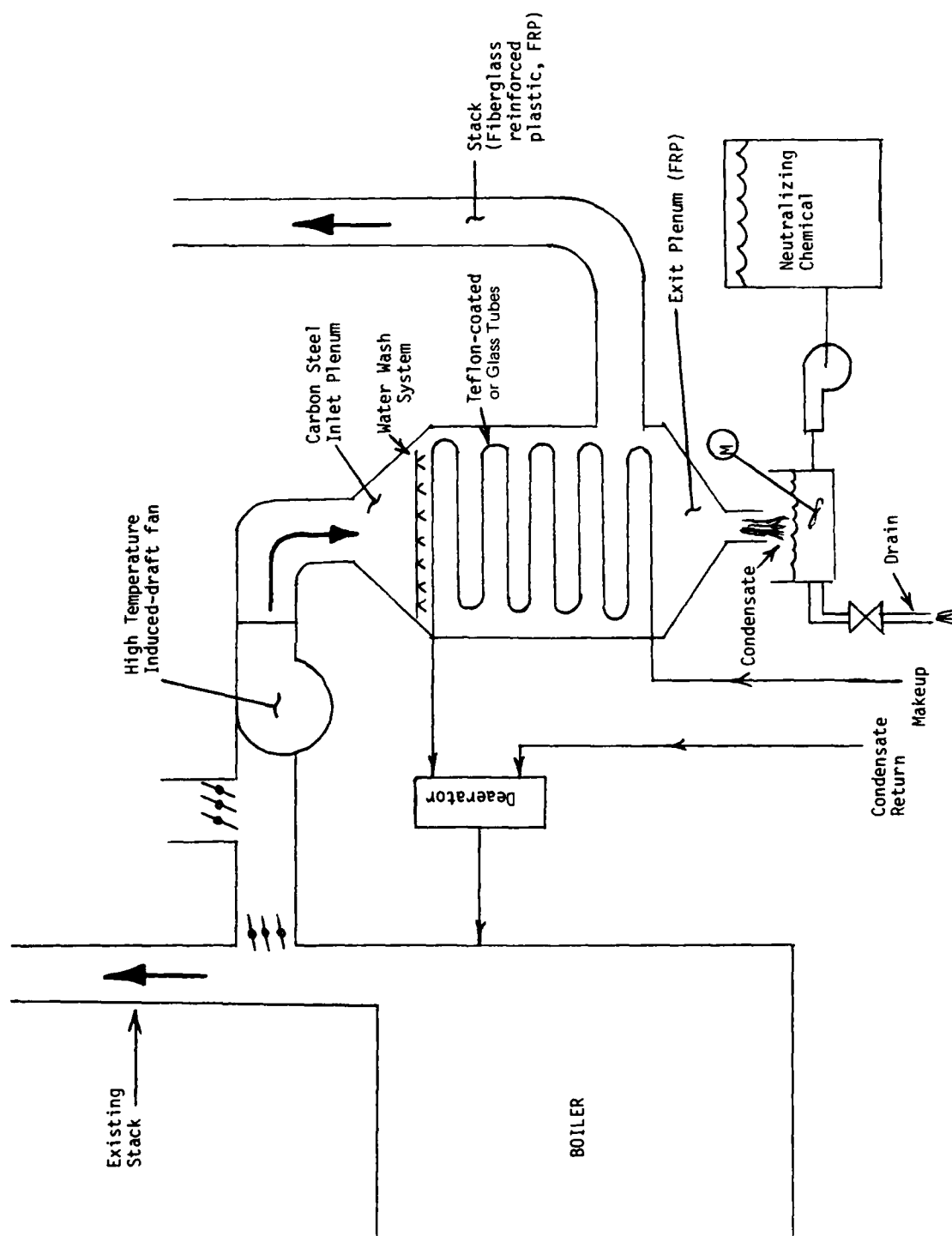
Figure 30. Schematic of an indirect-contact condensing heat recovery system.

83
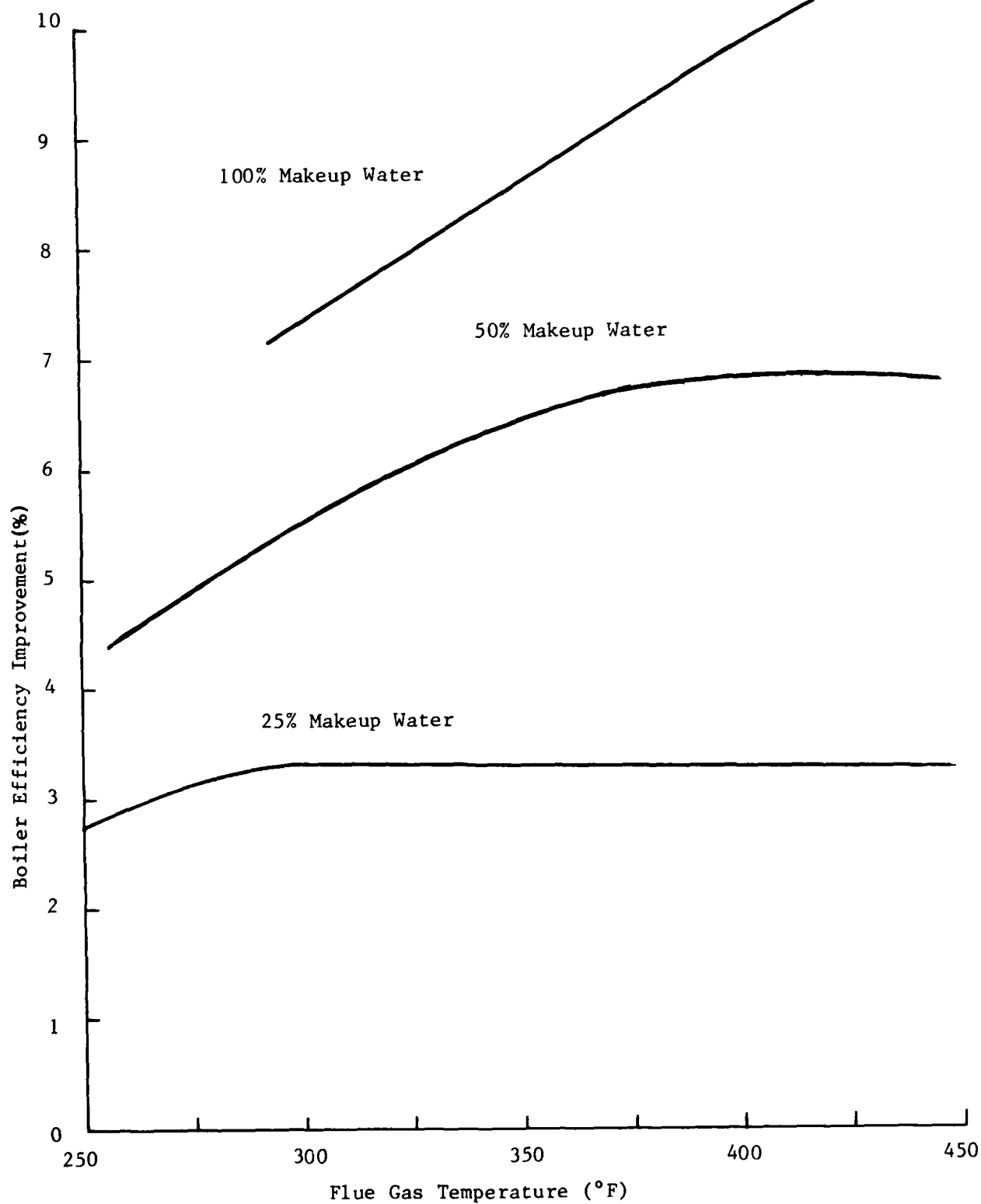
Figure 31. Boiler efficiency improvement resulting from an indirect-contact condensing heat exchanger.
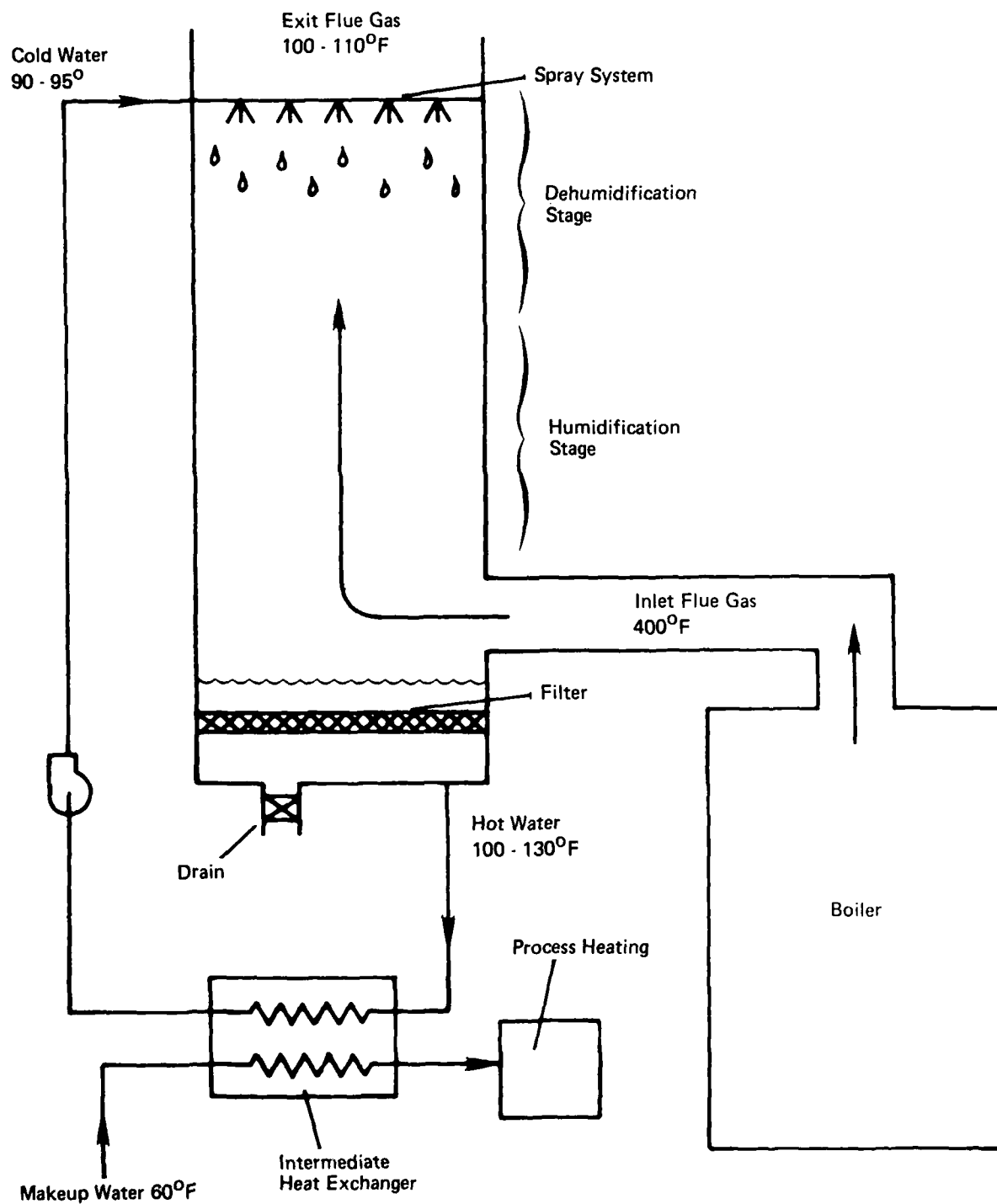
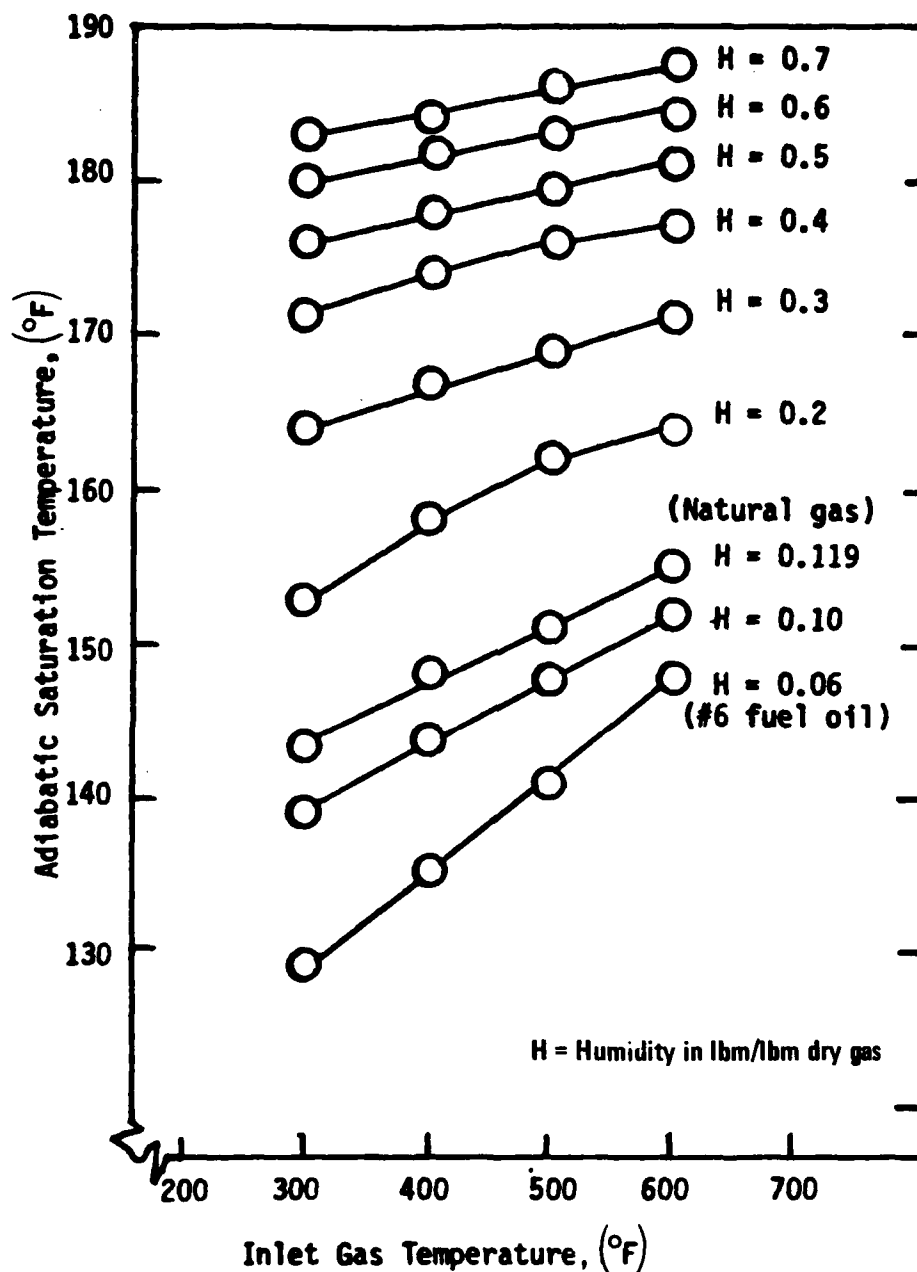Figure 32. Schematic of a direct-contact heat exchanger.

85

Figure 33. Dependence of adiabatic saturation temperature on inlet gas temperature and humidity for a direct-contact heat exchanger.
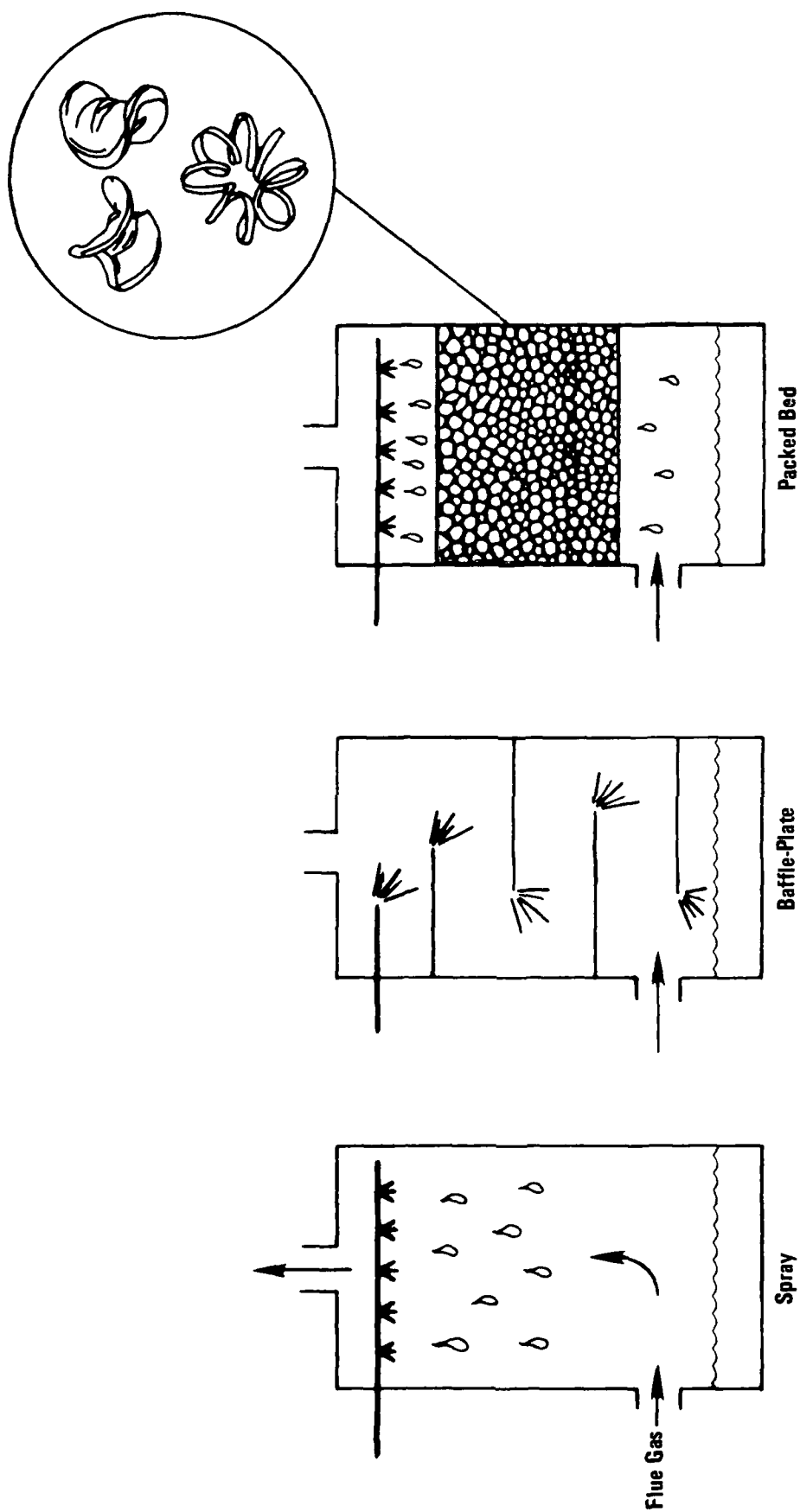
Figure 34. Direct-contact heat exchanger schemes.

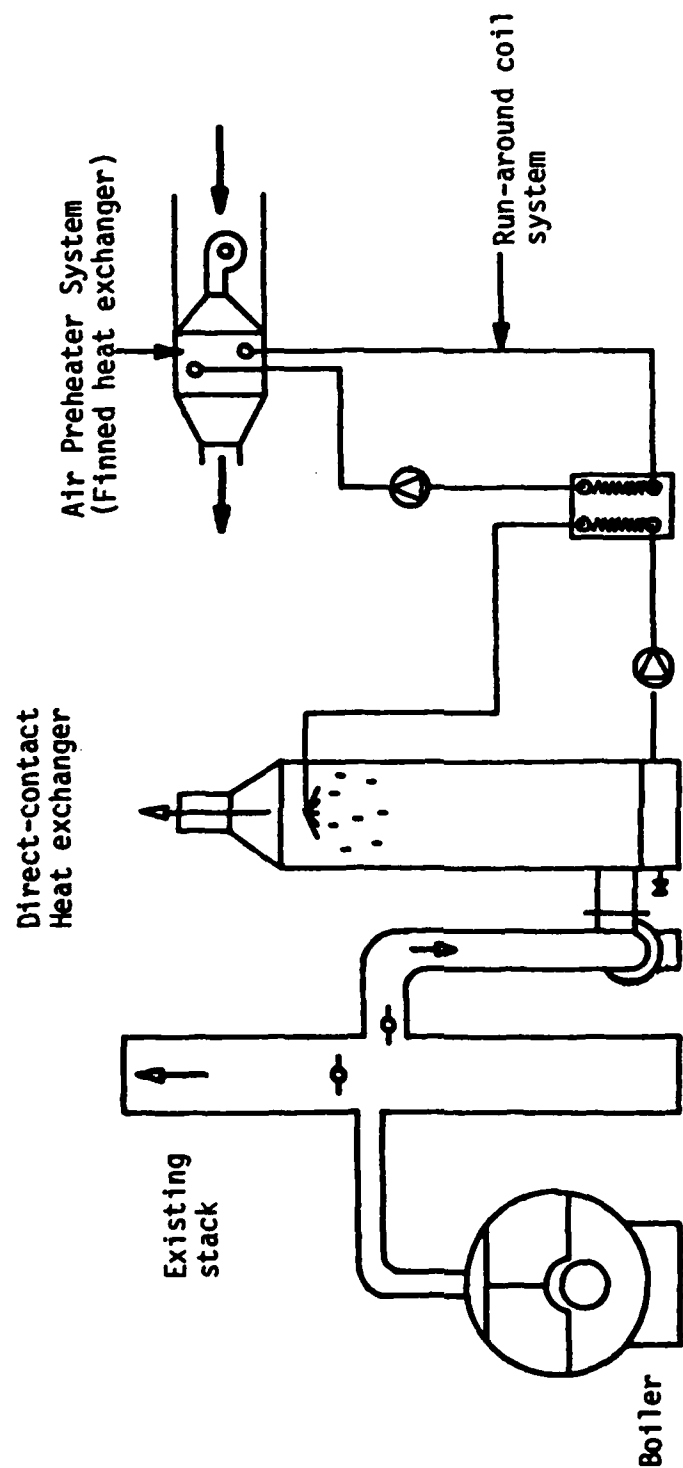Packed Bed

Baffle-Plate

Spray

Flue Gas

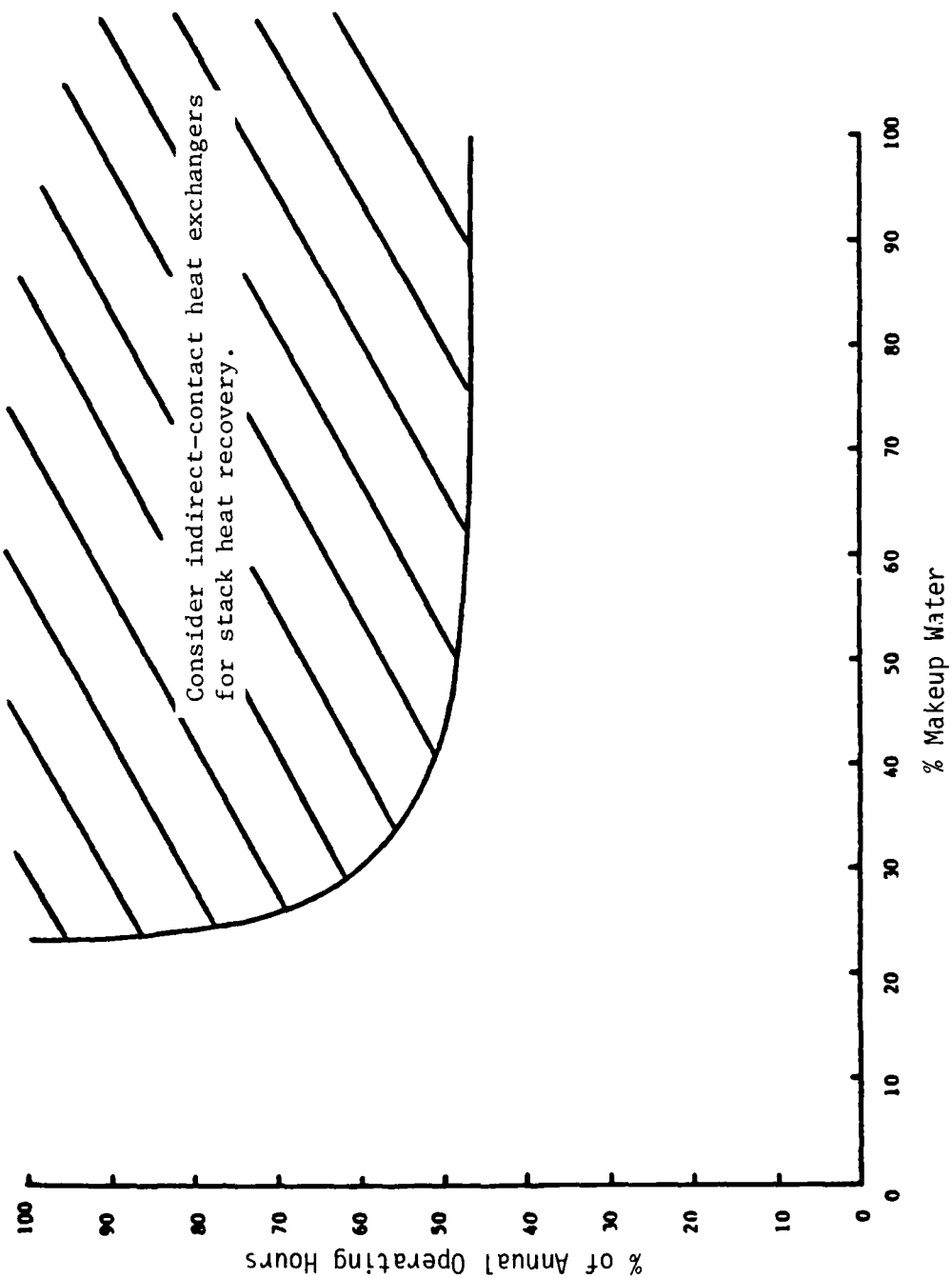Figure 35. Application of a direct-contact heat exchanger for building air heating.

Figure 36. Operating conditions for cost-effective installation of indirect-contact heat exchangers.

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32 Alternate energy source (geothermal power, photovoltaic power systems, solar systems, wind systems, energy storage systems)
33 Site data and systems integration (energy resource data, energy consumption data, integrating energy systems)
34 ENVIRONMENTAL PROTECTION
35 Solid waste management
36 Hazardous/toxic materials management
37 Wastewater management and sanitary engineering
38 Oil pollution removal and recovery
39 Air pollution
44 OCEAN ENGINEERING
45 Seafloor soils and foundations
46 Seafloor construction systems and operations (including diver and manipulator tools)
47 Undersea structures and materials
48 Anchors and moorings
49 Undersea power systems, electromechanical cables, and connectors
50 Pressure vessel facilities
51 Physical environment (including site surveying)
52 Ocean-based concrete structures
53 Hyperbaric chambers
54 Undersea cable dynamics

TYPES OF DOCUMENTS

85 Techdata Sheets   86 Technical Reports and Technical Notes   82 NCEL Guide & Updates   () None –
83 Table of Contents & Index to TDS   91 Physical Security   remove my name